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POWER EFFICIENT HYDRAULIC SYSTEMS

Volume I

STUDY PHASE

Rockwell International
North American Aircraft
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Energy saving concepts for aircraft hydraulic systems were studied in a two-phase program. Task I was an investigation of methods and techniques to reduce overall hydraulic system power requirements by lowering system demands and increasing component efficiencies. Task II involved hardware demonstration tests on selected concepts.

Task I: Study Phase. A baseline hydraulic system for an advanced aircraft design was established. Twenty energy saving techniques were studied as candidates for application to the baseline vehicle. A global systems analysis approach was employed. The candidates were compared on the basis of total fuel consumption and six qualitative factors. Nine of the most promising techniques were applied to a "Target System". The target system had a 28% reduction in energy consumption and an 868 lb weight reduction over the baseline aircraft. The study made one conclusion clear: Don't add weight to save energy.

Task II: Hardware Demonstration Phase. Two techniques demonstrated for energy savings

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were control valves with overlap and dual pressure level systems. Tests were conducted on control valves, a servo actuator, dual pressure pumps, and a lightweight hydraulic system simulator. Valves with 0.002 in. overlap reduced system energy consumption 18% compared to using valves with zero lap. Operation at 4000 psi reduced system energy consumption 53% compared to operation at 8000 psi. Pressure level switching was accomplished with excellent results.

EXECUTIVE SUMMARY

1.0 PURPOSE OF THE PROGRAM

The power requirements for military aircraft hydraulic systems have risen steadily from a few horsepower during World War II to over a thousand horsepower for the B-1B bomber. Significant increases in hydraulic power are projected for future Naval aircraft due to the growing number of control functions utilizing hydraulic power and requirements for higher control surface rates. The addition of engine and vectored thrust controls drives hydraulic power requirements up by 50 to 100 percent. Reduced static stability and higher maneuverability requirements of advanced aircraft necessitate higher surface rates. More efficient hydraulic systems have become increasingly important. High efficiency hydraulic systems must have minimum weight and minimum power extraction from the engines. Peak output power demands must more closely match system load requirements.

The purpose of this program is to investigate methods and techniques to reduce overall hydraulic system power requirements by lowering system demands and increasing component efficiencies. Twenty candidate energy saving concepts were studied for application to Naval aircraft in the 1990's time frame.

2.0 BENEFITS TO THE NAVY

This program was conducted to provide the Navy with a means of improving aircraft performance through the use of power efficient hydraulic systems. Smaller, lighterweight, more efficient hydraulic systems require less fuel. This translates into higher payloads, longer range, and improved aircraft performance. The program reviewed all known techniques and methods having a potential for saving energy or reducing power extraction. Concepts with the greatest potential have been identified. A basis has therefore been established for directing future effort into the most promising areas.

3.0 PROGRAM PLAN

An overview of the program is given in Figure 1. The program consisted of two major tasks:

Task I Study of energy saving techniques.
Task II Hardware demonstration tests on selected techniques.

Task I is reported herein (Volume I). Task II is reported in a separate document (Volume II).

4.0 STUDY PHASE SUMMARY (TASK I)

A global study approach was adopted which allowed quantitative comparison of the energy savings of each candidate technique on a total system basis. Fuel consumption was the common parameter used for comparison. Major subtasks of the study phase are listed in Figure 2. A baseline vehicle and baseline hydraulic system were established upon which the energy saving techniques were applied. The baseline vehicle is depicted in Figure 3. Results are somewhat dependant upon the vehicle; for example, weight is more critical in a fighter aircraft than in a transport. A methodology was then established which enabled quantitative comparison of changes in the hydraulic system; for example, leakage in a servo valve, weight of hydraulic tubing, or heat rejection in a pump. The energy saving techniques listed in Figure 4 were studied for application to the baseline hydraulic system. Each energy saving technique was applied to the baseline hydraulic system and total fuel savings were computed and compared to the baseline. A qualitative assessment of Reliability and Maintainability, Life Cycle Cost, Development Risk, Performance and Safety was made by a panel of subject matter experts. The most promising concepts are listed in Table 1.

POWER EFFICIENT HYDRAULICS

OBJECTIVES

- Reduce Overall Hydraulic System Power Requirements
- Increase Efficiency of Hydraulic Power Systems

TASK I—STUDY PHASE

- Establish Baseline Hydraulic System
- Analyze Proposed Energy Saving Techniques
- Comparative Analysis
- Establish Energy Efficient Target System

TASK II—HARDWARE DEMONSTRATION PHASE

- Design, Manufacture, and Assemble Selected Techiques Into Simulation System
- Demonstrate Selected Energy Efficient Techniques

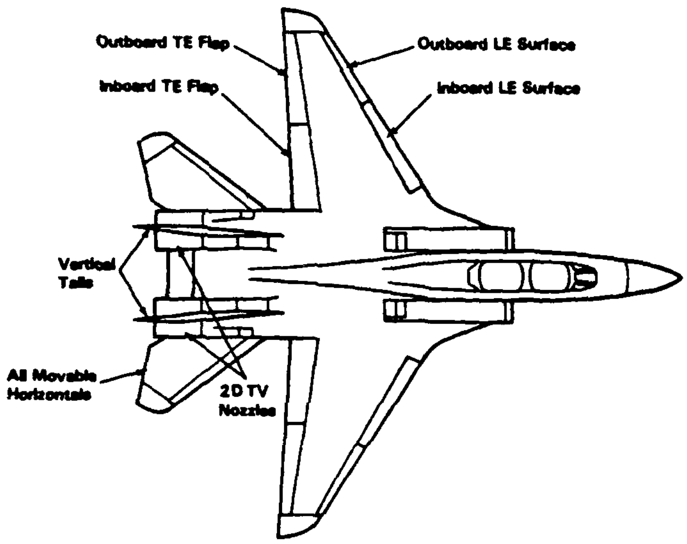
FIGURE 1. Program overview

ENERGY EFFICIENT STUDY

- Baseline System Definition
- Study Methodology
- Energy Saving Techiques
- Comparative Analysis
- Target System

FIGURE 2. Subtasks of Phase I study

BASELINE VEHICLE



GENERAL DESCRIPTION	
Gross Weight	64,000 Lb
Span	50 Ft
Length	70 Ft
T/W	0.9
Fuel	21,000 Lb
Stores	6,400 Lb
Wing Area	675 Ft^2

PERFORMANCE PARAMETERS

- $Mach_{Max}$ 1.8
- N_Z 6.5
- $N_{Z_{ULT}}$ 9.75
- V_{APP} 120 Knots
- Sink Rate 24 FPS

CONTROL EFFECTORS

Pitch	Roll	Yaw
• Horizontals	• Outboard TE	• Rudders
• Inboard TE	• Horizontals	• Vectored Thrust
• Vectored Thrust	• Vectored Thrust	

NA 4877C

FIGURE 3. Baseline vehicle

ENERGY SAVING TECHNIQUES

- Pumps and IAPS
- Distribution System
- Accumulators
- Advanced Actuation
  - Variable Displacement
  - Slimline
  - Pressure Intensifiers
- Control Valves
  - Aiding Load Recovery
  - Flow Augment
  - Nonlinear Valves
- Multipressure System
- Hybrid Hyd/Em
- Advanced Materials
- Design Margins
- Thrust Vectoring
  - Trim T/V
  - Hot Gas Diverters
- Vehicle Control System
  - Command Optimization
  - Variable Gain/Bandwidth

FIGURE 4. Energy saving techniques studied

TABLE 1. Energy saving concepts

CANDIDATE CONCEPTS	RATING	
Advanced Materials	1.30	(best)
Dual-Pressure System	0.41	
Pumps	0.37	
Non-Linear Valves	0.36	
Distribution System	0.34	
Variable Gain/Bandwidth	0.19	
Accumulators	0.14	
Hybrid Hydraulics/Electro-mechanical	0.07	
Flow Augmentation	0/+0.50	

A "Target Hydraulic System" was designed using the most promising techniques and fuel consumption and weight were analyzed. The target hydraulic system achieved a 28% reduction in energy consumption. This is equivalent to an 868 lb weight savings in the baseline vehicle.

The study made one conclusion clear:

"Don't Add Weight to Save Energy".

Weight is dominate as the largest energy consumer. Even a fraction of a pound added by an "energy saving" device typically resulted in much greater fuel consumption than the energy saved by the device.

PREFACE

This report documents an investigative program conducted by Rockwell International Corporation, North American Aircraft Operations, Columbus, Ohio, under Contract N62269-85-C-0259 with the Naval Air Development Center, Warminster, Pennsylvania. Technical direction was administered by Mr. J. Ohlson and Mr. D. Bagwell, Materials Application Branch, Aircraft and Crew Systems Technology Directorate, Naval Air Development Center (6061).

This report presents the results of a two-phase program to study and demonstrate methods and techniques to improve the operating efficiency of hydraulic systems in advanced Naval aircraft. This work is related to tasks performed under the Lightweight Hydraulic System Development Program, Contract N62269-80-C-0261. The report consists of two volumes:

Volume I	Study Phase
Volume II	Hardware Demonstration Phase

The project engineer for the Power Efficient Hydraulic Systems program was Mr. W. Bickel . Acknowledgment is given to the following engineers for their contributions to this report:

W. Andrews	Control Systems
J. Denniston	Air Vehicle
B. Holland	Hydraulic Systems
E. Kauffman	Reliability and Maintainability
L. Kohnhorst	Control Systems

Appreciation is extended to the many individuals who provided helpful support and constructive criticism of the program; in particular, Mr. J. Ohlson and Mr. D. Bagwell of the Naval Air Development Center, and Mr. L. Biafore and Mr. A. Eckles of Rockwell International.

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1.0 INTRODUCTION

1.1 BACKGROUND

The power requirements for hydraulic systems in military aircraft have risen steadily from a few horsepower during World War II to over 1000 hp in the B-1B bomber. Significant increases in hydraulic power are projected for future Naval aircraft due to the growing number of control functions utilizing hydraulic power and requirements for higher control surface rates. The addition of engine and vectored thrust controls drives hydraulic power requirements up by 50 to 100%. Reduced static stability and higher maneuverability requirements of advanced aircraft necessitate higher surface rates. More efficient hydraulic system designs which minimize power consumption, weight, and volume, become increasingly important as power extraction increases.

1.2 PROGRAM OBJECTIVES

The program objectives were to investigate methods and techniques to reduce overall hydraulic system power requirements by lowering system demands and increasing component efficiencies. Results of the study were to be applied to a baseline advanced Naval aircraft design to establish the total energy saving potential of a hydraulic system with minimum weight, minimum power extraction from the engines, and with peak output power demands matched to system load requirements. Laboratory tests were then to be conducted on specially designed hardware to demonstrate selected energy saving techniques.

1.3 SCOPE OF WORK

The program was performed in two phases:

Task I Study Phase

Task II Hardware Demonstration Phase

Task I consisted of the following:

- o Determination of study methodology
- o Definition of baseline vehicle
- o Establishment of baseline hydraulic system
- o Evaluation of candidate energy saving techniques

  Pumps
  Integrated actuator packages
  Distribution system (5 approaches)
  Accumulators
  Variable displacement actuators
  Slimline actuators
  Pressure intensifiers
  Non-linear control valves
  High-overlap control valves
  Aiding load recovery valves
  Flow augmentation valves
  Multipressure systems
  Hybrid hydraulic/electro-mechanical systems
  Advanced materials
  Design margins
  Hot gas diverters
  Trim thrust vectoring
  Variable gain/bandwidth
  Command optimization

- o Apply most promising techniques to target system
- o Determine weight and energy savings of target system over the baseline.

Task II consisted of the following:

- o Design test parts

  Actuator modification

  Test fixture modification

- o Procure demonstration hardware

  Dual pressure pumps

  Direct drive control valves and electronics

- o Conduct demonstration tests
- o Analyze test results.

Task I is presented herein (Volume I). Task II is presented in a separate document (Volume II).

2.0 STUDY PHASE (TASK I)

2.1 STUDY METHODOLOGY

The analysis approach, summarized in Figure 1, was developed around two fundamental criteria: 1) evaluation and comparison of energy saving techniques must be based upon total aircraft energy consumption -- not merely upon individual components; and 2) the evaluation and comparison must be based upon realistic usage -- not upon maximum or ideal conditions which are seldom encountered in practice. Therefore, a global or total systems approach is required. For example, it is not productive to reduce hydraulic system energy losses by replacing a hydraulic component or subsystem with a non-hydraulic one that is more efficient but heavier.

The study approach takes into account all energy losses and inefficiencies -- both direct and indirect. Direct energy losses are associated with the efficiency of system components. Examples are internal leakage in control valves and pumps, pressure drop in hydraulic tubing, and friction in actuators. Indirect energy losses are associated with weight and/or size effects. If one component weighs more than another (which performs the same function), the heavier weight results in more energy consumption. To support the additional weight, aircraft angle-of-attack must be increased which in turn induces more drag. Additional engine thrust is then necessary to offset the higher drag and maintain airspeed, thus raising energy consumption. Size can also increase energy consumption if it requires enlarging the aircraft moldline which in turn adds drag.

System element efficiencies must be accounted for in comparing one energy saving technique with another. For example, it can be seen from Figure 2 that the energy loss of a component must be divided by the efficiencies of all system elements upstream of the component to obtain the total loss resulting from that component. In order to compare the total impact of changes in

ANALYSIS APPROACH

STUDY CRITERIA

- **Based Upon Total Aircraft Energy Consumption**
- **Based Upon Usage (Duty Cycles)**

GLOBAL APPROACH

- **Direct Energy Components**
- **Indirect Energy Components**
- **System Efficiencies**

ENERGY CONSUMPTION IS COMPARED IN TERMS OF LBS FUEL/AIRCRAFT LIFE

Figure 1. Analysis approach

ENERGY EFFICIENCY BLOCK DIAGRAM

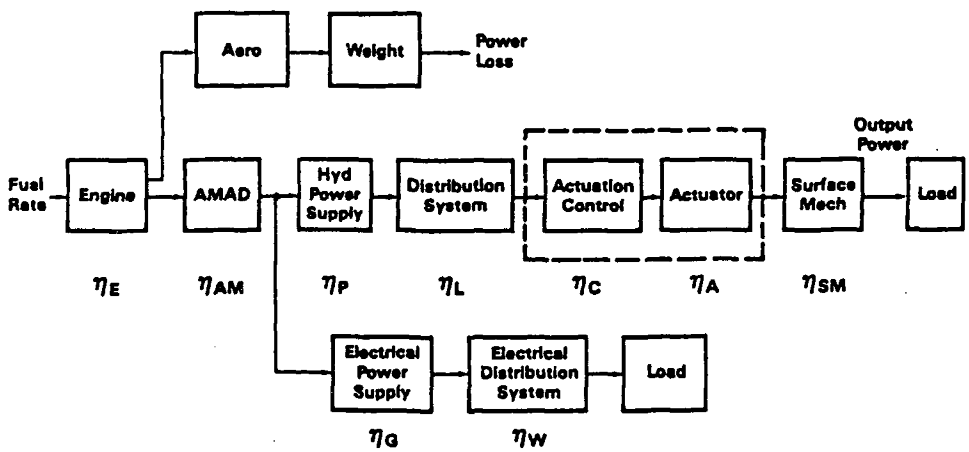


Figure 2. Energy efficiency block diagram

different areas of the hydraulic system, element efficiencies must be determined and comparisons made at a common reference point. The common reference point chosen for this study was aircraft fuel. Fuel is stored energy and fuel usage rate is the equivalent of power. Total fuel consumption over the life of the aircraft was therefore established as the basis for comparing the candidate energy saving techniques and is expressed in units of M-lb (millions of pounds) of fuel.

Application of this approach requires the following:

1. Definition of the baseline vehicle and hydraulic system in sufficient detail to perform the required analysis.
2. Definition of a composite mission for the baseline vehicle.
3. Development of an engine/aero model for computing fuel consumption rate due to primary/secondary flight controls and utility functions.
4. Definition of actuation usage.
5. Establishment of system/component efficiencies.
6. Computation of energy consumption and losses for all direct and indirect components, computation of fuel consumption, and summation to determine total fuel consumption.
7. Qualitative assessment of the energy saving techniques since conclusions and recommendations can not be made upon energy savings alone.
8. Comparative analysis of specific energy savings methods and techniques.

One purpose of the study was to evaluate and compare many energy savings techniques, determine those with the most potential, then focus on the most promising candidates. It was necessary to limit the depth of the study because of funding constraints.

Item 1 above (Baseline Vehicle and Baseline Hydraulic System) is discussed in sections 2.2 and 2.3; Items 2 through 4 are discussed in this section (2.1); Items 5 and 6 are discussed in section 2.4; and Items 7 and 8 are presented in section 2.5.

2.1.1 Aircraft Mission

The multi-mission attack aircraft chosen for this study is typical of current projections for next generation Naval aircraft. The data base for this aircraft evolved from ATA and VFMX studies conducted at Rockwell. The two basic missions, Air-to-Air and Air-to-Surface, are outlined in Tables 1 and 2. From these prior studies a composite mission was established upon which the analysis was based. The composite mission is shown in Table 3, and is representative of an attack encounter. The composite mission duration is 162 minutes and is divided into seven legs. Flight conditions were established for each leg and, for purposes of this analysis, held constant throughout the leg.

2.1.2 Aero/Engine Model

An aero/engine model, developed by Rockwell in previous studies, established the mathematical relationship between fuel consumption, weight, and engine shaft power extraction. These relationships are expressed in terms of fuel consumption rate coefficients. The fuel consumption rate per horsepower coefficient is in units of lb(fuel)/hr/hp and the fuel consumption rate per pound weight is in units of lb(fuel)/hr/lb(weight).

This approach is generic and can be applied to any vehicle, however coefficient values are dependent upon the aircraft lift-to-drag ratio, engine performance, and flight conditions. Lift-to-drag ratio curves were reviewed for a number of advanced aircraft which fit the multi-mission role and found to be quite similar. Representative drag polars established for the baseline vehicle are shown on Figure 3.

TABLE 1. Air-to-air missions

MISSION	DESIRED PERFORMANCE
• FLEET AIR DEFENSE • COMBAT AIR PATROL/OAB • DEFENSE AGAINST ESCORTED BACKFIRES	• 300 NM CRUISE • 100 NM DASH @ 1.5 M (ONE WAY) • 2-3 HR LOITER • 2-MIN. COMBAT 1.5 M @ 35,000 FT
• DECK LAUNCHED INTERCEPT	• 300 NM DASH @ 1.5 M (SECONDARY MISSION)
• FIGHTER ESCORT	• 750 NM CRUISE • 40 NM DASH @ 10,000 FT (ONE WAY) • 10-MIN. COMBAT @ 10,000 FT IRT

TABLE 2. Air-to-surface missions

MISSION	DESIRED PERFORMANCE
• INTERDICTION • SCATTERED TARGETS • STRONG SCATTERED DEFENSES • SURFACE COMBATANT STRIKE/ SURFACE SURVEILLANCE/TARGETING • FLEET ATTACK • LOW LEVEL TACTICS	• 750 NM CRUISE • 50 NM DASH - 0.9 M @ S.L. • 5-MIN. COMBAT @ S.L. IRT
• DEFENSE SUPPRESSION • HARM ESCORT OF ASUW & INTERDICTION AIRCRAFT	• 750 NM CRUISE • 40 NM DASH - 0.9 M @ S.L. • 8-MIN. COMBAT @ S.L. IRT
• LONG RANGE STRIKE • LOW LEVEL PENETRATION • HEAVILY DEFENDED TARGETS	• 700 NM CRUISE • 100 NM DASH - 0.9 M @ S.L. • 5-MIN. COMBAT @ S.L. IRT
• MINE LAYING	• FALL-OUT PERFORMANCE (SECONDARY MISSION)

TABLE 3. Composite mission

MISSION LEG	MODE	DURATION, MIN.	PERCENT OF MISSION TIME	ALTITUDE, FT.	MACH NO.
1	Takeoff	3	1.9	S.L.	0.28
2	Climb and Cruise	48	29.6	35K	0.8
3	Loiter and Descent	36	22.2	30K	0.7
4	Dash	4	2.4	S.L.	1.1
5	Combat	5	3.2	10K	0.6
6	Cruise and Descent	48	29.6	40K	0.8
7	Landing	18	11.1	S.L.	0.28
		162	100.0%		

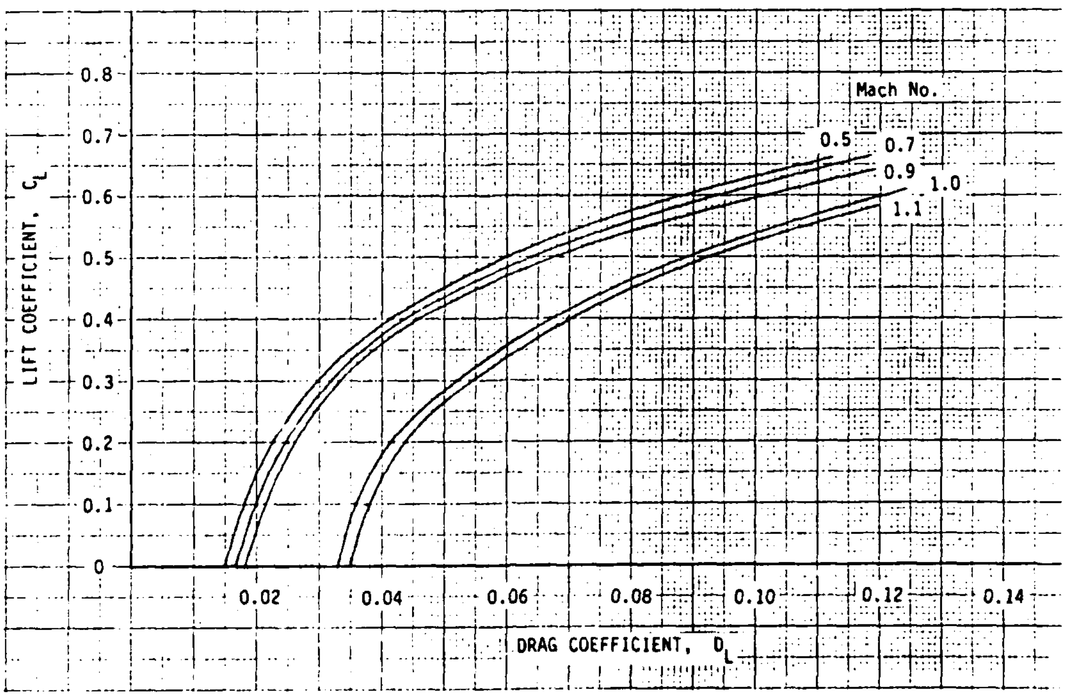


Figure 3. Baseline vehicle drag polars

Engine performance was based on General Electric F110 engine data generated by computer models employed in advanced studies. Operating conditions for the composite mission legs are shown in Table 3.

As a starting point in the derivation of the fuel consumption coefficient for weight, the average aircraft weight was estimated for each mission leg. Fuel usage was computed from the engine model for the leg conditions and this average weight. The average mission leg weight was then revised by the weight of the fuel consumed and the computations were repeated. Iteratively, fuel consumption was established for each leg in turn. Average fuel consumption rate coefficients were then calculated for each leg. Flight condition data used for the calculations is shown in Tables 4 and 5. Table 4 defines air density and speed for each mission leg. Table 5 lists lift and drag data for each leg based upon the drag polars in Figure 3. Calculations for all mission legs except takeoff were based upon equilibrium for the average weight during the leg; that is, thrust is equal to drag and lift is equal to weight. For takeoff, thrust was established at a value greater than drag to produce the necessary acceleration. Values for drag coefficient (C_D) and lift drag slope (C_L/C_D) were then determined from Figure 3 and are presented in Table 5. The calculations involved the following equations:

$$L = W_T = \frac{1}{2} \rho V^2 S C_L$$

or,

$$C_L = \frac{2 W_T}{\rho V^2 S}$$

$$C_D = f(C_L, M_N)$$

$$D = T_H = \frac{1}{2} \rho V^2 S C_D$$

where,

C_D = drag coefficient
C_L = lift coefficient
D = drag
M_N = Mach number
T_H = thrust
S = total wing area
V = velocity
W_T = weight
ρ = air density

TABLE 4. Calculation values

MISSION LEG	AIR DENSITY, $LB\text{-}SEC^2/FT^4$	SPEED (FT/SEC)	SPEED (KNOTS)
1	0.00238	300	180
2	0.000694	778	461
3	0.000891	696	412
4	0.00238	1300	770
5	0.00230	645	370
6	0.000587	775	460
7	0.00238	300	180

TABLE 5. Aircraft lift/drag data

MISSION LEG	C_L	C_D	SLOPE C_L/C_D	DRAG (LB)
1	0.86	0.16	2.7	11560
2	0.42	.047	6.25	6870
3	0.37	.036	8.9	5240
4	0.04	.034	22	46100
5	0.16	.02	15.2	6458
6	0.35	.038	8.0	4521
7	0.54	.072	3.5	5205

For equilibrium, drag must equal thrust. This thrust value (see Table 5) was then used in the G.E. F110 engine model to obtain fuel consumption rate and specific fuel consumption (SFC). Engine performance data, based on the F110 engine model is summarized in Table 6. Column 2 of this table shows the thrust per engine, Column 3 shows fuel usage per engine and Column 4 shows specific fuel consumption per engine. Total fuel consumption for two engines is given in Column 5 for the mission leg. This is computed by multiplying mission leg time by the fuel consumption rate for two engines. A summary is shown in Table 7. The average weights were calculated based on fuel usage and the dropping of 6400 lb of stores in mission leg 5. Fuel weight is the average value of fuel onboard at the start and the end of each leg.

2.1.2.1 Effect of Weight on Fuel Consumption. The variation in fuel usage caused by changes in aircraft system weight can be derived from the data established in Tables 5 and 6. This relationship is developed for each mission leg and the total mission.

The lift-to-drag ($L/_D$) ratio is equal to the C_L/C_D ratio. This can be seen from the lift and drag equations, i.e.:

$$\frac{L}{D} = \frac{\frac{1}{2}\rho V^2 S C_L}{\frac{1}{2}\rho V^2 S C_D}$$

$$= \frac{C_L}{C_D}$$

TABLE 6. Fuel usage

MISSION LEG	NET THRUST PER ENGINE (LB)	FUEL USAGE PER ENGINE (LB/HOUR)	SFC LB(fuel)/Hr LB(thrust)	TOTAL FUEL USAGE (LB)
1	12000	9000	0.785	900
2	3400	2800	.88	4480
3	2620	1980	.98	2380
4	23000	51000	3.0	6630
5	3200	2800	0.96	470
6	2300	1900	0.90	3040
7	2600	2500	0.98	1500

TABLE 7. Mission summary

MISSION LEG	FUEL REMAINING (LB)	WEIGHT AT END OF LEG (LB)	AVERAGE WEIGHT (LB)
INITIAL	21000	64000	-
1	20100	63100	63550
2	15620	58620	60860
3	13240	56240	57430
4	6610	49610	52920
5	6140	42740	46170
6	3100	39700	41220
7	1600	38200	39700

For equilibrium, lift must equal weight and drag must equal thrust; therefore, the incremental change in thrust for an incremental change in drag can be computed from the slope (C_L / C_D) of the drag polar, i.e.,

$$\Delta T \triangleq \Delta D = \Delta L \times \left(\frac{\Delta C_D}{\Delta C_L}\right)$$

$$= \Delta W_T \left(\frac{\Delta C_D}{\Delta C_L}\right)$$

Specific fuel consumption (SFC) is a performance parameter of jet engines which relates fuel flow rate to thrust; i.e.,

$$SFC \triangleq \frac{\text{FUEL FLOW RATE (lb/HR)}}{\text{THRUST (LB)}}$$

The relationship between fuel consumption rate and aircraft weight can be found by combining the two foregoing equations; i.e.,

$$\Delta T = \Delta W_T \left(\frac{\Delta C_D}{\Delta C_L}\right) \approx \frac{\Delta \text{FUEL FLOW RATE}}{SFC}$$

$$\text{OR,} \quad FCR_{LB} \triangleq \frac{\Delta \text{FUEL FLOW RATE}}{\Delta W_T} \approx SFC \left(\frac{\Delta C_D}{\Delta C_L}\right)$$

The incremental fuel consumption rate per pound (FCR_{lb}) for the average leg conditions are listed in Table 8. A composite value was computed which is an average weighted by the percent of time spent in each mission leg. This was computed by:

$$\text{Composite Value} = \sum_{i=1}^{7} \frac{FCR(i)T_L(i)}{T_{M_{TOTAL}}}$$

2.1.2.2 Shaft Horsepower Extraction. The effects of shaft horsepower extraction on engine performance is more difficult to obtain. It is affected by a number of variables which include the engine operating point, flight condition, magnitude of shaft horsepower variation, and effects of engine bleed air to name a few. The gear box design, including starting and redundancy features, also affects extraction efficiency. A computer model of the engine was run which shows the effects of different levels of horsepower extraction on engine specific fuel consumption (SFC). Since this was a modification of the normal engine model, the data were only run at one point having low thrust values and low SFC values so the effects of horsepower extraction are more apparent. The data are plotted for 125, 300 and 475 horsepower extraction loads on the engine in Figure 4 and tabulated in Table 9. The SFC shown in Table 9 is the difference between the 475 hp curve and the 125 hp curve at the specific operating condition. Fuel rate was calculated from

$$\text{Fuel Rate} = \text{Thrust} \times \text{SFC}$$

$$\text{or} \quad \text{Fuel Rate} = \text{lb} \times \frac{\text{lb/hr}}{\text{lb}} = \text{lb/hr}$$

The only difference between the two curves is an additional 350 hp shaft extraction. If fuel rate is divided by 350 hp, a value of fuel consumption rate per horsepower is obtained (Column 5).

TABLE 8. Fuel usage variations with weight

MISSION LEG	SLOPE, C_L/C_D	SFC, $\left(\frac{LB/HR}{LB}\right)$	$\frac{1}{FCR_{LB}}$, $\left(\frac{LB}{LB/HR}\right)$	FCR_{LB}, $\left(\frac{(LB\ FUEL)/HR}{LB\ (WT)}\right)$	TIME RATIO, $\left(\frac{T_L}{T_{TOT}}\right)$
1	2.7	.785	3.44	0.29	0.019
2	6.25	.88	7.1	0.14	0.296
3	8.9	.98	9.08	0.11	0.222
4	22	3.0	7.33	0.14	0.024
5	15.2	.96	15.83	0.06	0.032
6	8.0	.90	8.89	0.11	0.296
7	3.5	.98	3.57	0.28	0.111
COMPOSITE VALUE	7.65	0.97	7.89	0.14	

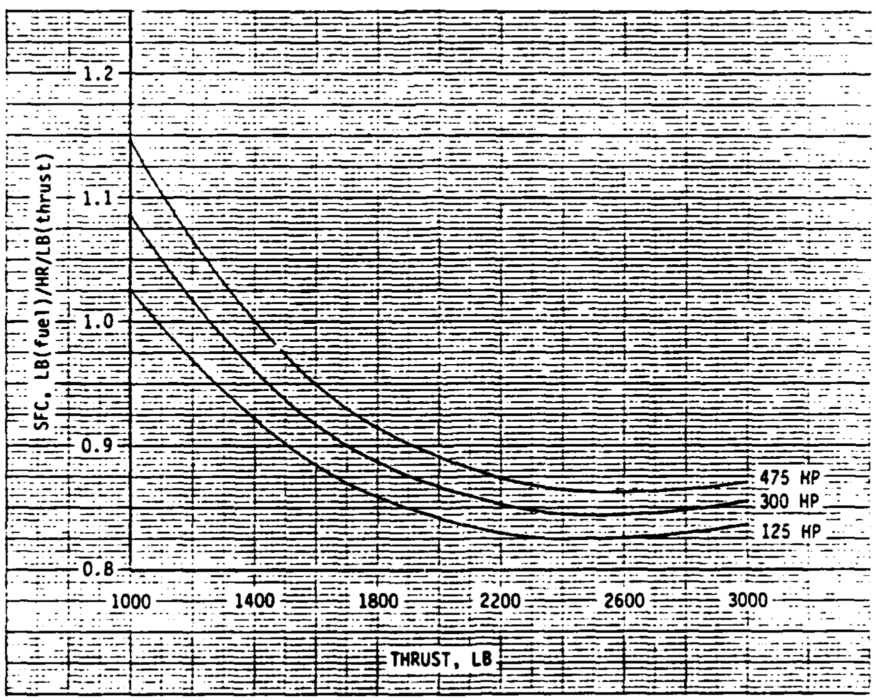


Figure 4. Engine data illustrating effects of shaft horsepower extraction

TABLE 9. Shaft extraction fuel consumption rat
vs. thrust level

THRUST	ΔSFC $\left(\frac{LB/HR}{LB}\right)$	MID SFC $\left(\frac{LB/HR}{LB}\right)$	FUEL RATE (LB/HR)	FCR_{HP} $\left(\frac{LB/HR}{HP}\right)$
1000	0.122	1.085	122	.35
1400	0.078	.96	109.2	.31
1800	0.058	.885	104.4	.30
2200	0.044	.850	96.8	.28
2600	0.038	.845	96.8	.28
3000	0.033	.852	99	.28

TABLE 10. Shaft extraction fuel consumption rate vs. mi

MISSION LEG	SFC $\left(\frac{LB/HR}{LBS}\right)$	FCR_{HP} $\left(\frac{LB/HR}{HP}\right)$	TIME R
1	0.785	0.26	0.1
2	0.88	0.29	0.2
3	0.98	0.32	0.2
4	3.0	0.99	0.0
5	.96	0.32	0.0
6	.90	0.30	0.2
7	.98	0.32	0.1

Fuel consumption rate is directly related to engine efficiency which is reflected in the SFC number for the specific operating point. This is true for both shaft extraction and thrust. Engine efficiency can be taken into account by normalizing fuel consumption rate per horsepower extraction (FCR_{hp}) to the engine operating point (i.e., MID SFC). Column 6 is derived by dividing column 5 by column 3 (Table 9). It can be seen that the normalized values are approximately equal; an average value is 0.33. The fact that the normalized values of column 6 approach a common value shows that fuel consumption rate for horsepower extraction is directly related to engine efficiency as defined by SFC. The deviations in the values are within the reading accuracies of the plot in Figure 4. To minimize reading errors, the column values were averaged to obtain the normalized value (at SFC = 1) of lb/hr per hp. The normalized FCR_{hp} is 0.33 at an SFC of 1. This value was used to show the effects of horsepower extraction on the composite mission. This data is developed in Table 10. An overall value for the composite mission is obtained by multiplying each leg value by the leg time, summing the components and dividing by the total mission time. The composite mission value obtained is 0.32 lb/hr/hp at the engine shaft.

The relationship between fuel consumption rate and weight, and the relationship between fuel consumption rate and shaft power extraction have thus been developed and are described by two coefficients: Fuel consumption rate per pound weight and fuel consumption rate per horsepower shaft extraction. Coefficient values are dependent upon the specific engine, vehicle, and mission (operating point). This illustrates the necessity of defining specifically these parameters for the study.

The engines operate very inefficiently in the dash leg of the mission. The impact of the dash leg on composite mission values was investigated by eliminating the dash leg and by approximately doubling the dash leg time. The two fuel consumption coefficients for the modified missions are compared with the baseline mission in Table 11. The "increased dash" mission raises the FCR_{lb} coefficient by 14% and the FCR_{hp} coefficient by 9%.

TABLE 11. Impact of dash leg on composite mission values

ITEM	BASELINE MISSION	ZERO DASH MISSION	INCREASED DASH MISSION
Mission Times			
Total	2.9	4.16	1.36
Cruise	1.6	2.56	0.8
Loiter	0.6	1.16	-
Dash	0.065	-	0.12
FCR_{LB}	0.14	0.13	0.16
% Change	-	-7%	+14%
FCR_{HP}	0.33	0.30	0.36
% Change	-	-9%	+9%
NOTE: TOTAL FUEL HELD CONSTANT			

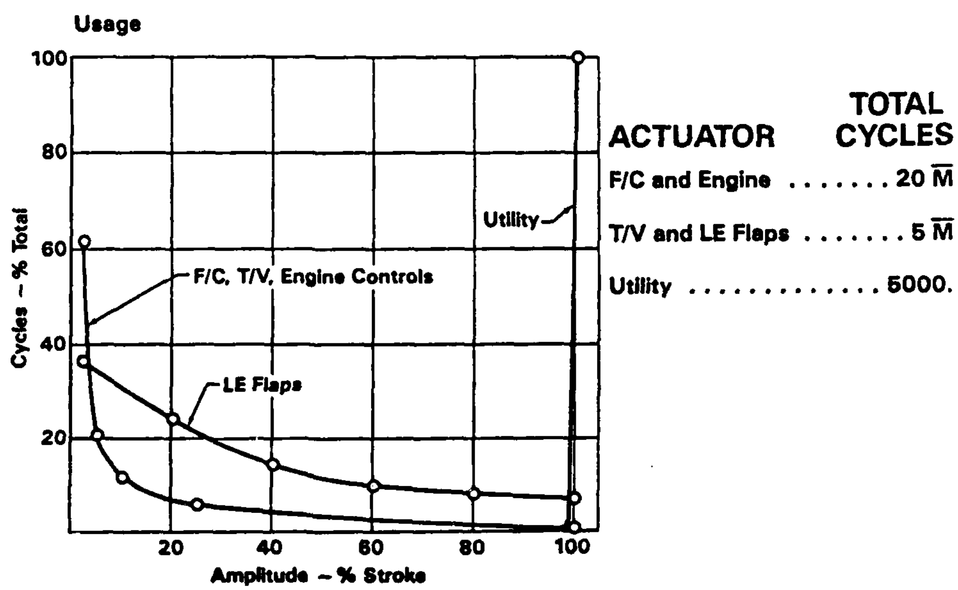


Figure 5. Actuator usage

2.1.3 Usage Functions and Efficiencies

To compute the total energy consumed by an actuator during the life of the aircraft, the total usage of the actuator must be determined. This involves estimating load, deflection amplitude, rate, and duty cycle based on mission requirements. The usage function defines the distribution of cycles in terms of amplitude. This information is difficult to ascertain for a new aircraft design. One measure that could be used is the endurance requirements specified in MIL-C-5503C. Later versions of this specification have deleted specific numeric requirements, leaving them instead to be detailed in the aircraft specification. Modern aircraft with control-by-wire (CBW) systems have significantly increased usage over non-CBW systems. The current design goal for advanced primary flight controls in reduced stability aircraft designed for extended life (10,000 hr) is 20 million cycles. The usage functions employed in this study are shown in Figure 5.

A high percentage of the 20 million cycles for flight controls are in the low amplitude region. Thrust vectoring and LE flap actuators are employed less frequently but generally experience greater deflection amplitudes. Thrust vectoring controls are used only when other controls are inadequate for the commanded maneuver such as during take-off and landing. Utility actuators are basically two position devices which operate through one full stroke cycle per flight.

Some energy saving techniques depend upon flow rate. To compute the energy consumption of these techniques, actuation rate must be known. This can be determined from the frequency of the usage cycles. Table 12 lists the frequency used in the analysis for each control function. Flight control in general is in response to vehicle disturbances which occur at the airframe natural frequency.

Direct lift, thrust control, and utility functions occur at relatively lower frequencies. For sinusoidal motion, the flow rate is related to frequency by:

$$Q = D_m \dot{\theta} = D_m A \omega$$

where,

Q = Flow Rate
D_m = Actuator displacement
$\dot{\theta}$ = Actuator rate
A = Amplitude
ω = Frequency

When amplitude (specified by the usage functions) and frequency (specified by Table 12) demand flow in excess of the valve design (no-load flow), frequency will be reduced as a result of rate limiting (i.e., $\omega = Q_{MX} \div D_m A$).

TABLE 12. Usage function frequencies

Control Function	Frequency, Hz
Longitudinal	25
Lateral	25
Directional	12
Direct Lift	6
Thrust Vectoring	25
Thrust Magnitude	6
Utility	3

2.1.4 Fuel Consumption Calculations

Fuel (energy) is the common parameter used to compare energy saving methods and techniques (concepts). The use of this parameter permits direct numeric comparison of all concepts regardless of the system element involved, takes into account all system efficiencies, and allows comparison of direct and indirect energy consumption components. Figure 6 depicts the calculations necessary to determine the total fuel consumption over the life of the aircraft due to hydraulic system power requirements.

An aircraft mounted accessory drive (AMAD) efficiency (η_{AM}) value of 0.9 was chosen as being typical of current equipment, Figure 6. The power growth factor represents the quantity of fuel necessary to transport the fuel required to supply the power. In other words, fuel (which has weight) must be carried to provide power and this results in an indirect fuel consumption component. (This factor is discussed in Section 2.1.4.1.) The fuel consumption coefficients (lb-fuel/hr/hp and lb-fuel/hr/lb-wt) are discussed in Section 2.1.2. Usage functions and aircraft life are discussed in Section 2.1.3. The weight growth factor accounts for the increase in aircraft weight (structure and fuel required to carry the additional structural weight) required by an increase in hydraulic system weight; this factor is discussed in the next section. Work/cycle is the work (energy) drawn from the hydraulic system to move an actuator through one complete cycle. Electrical power input is the power required by actuator direct drive valve torque motors and electronic drive units.

2.1.4.1 Weight and Power Growth Factors. The weight growth factor (WTGF) is defined as the ratio of change in gross take-off weight to change in equipment weight. The WTGF typically varies from 1.5 to 5 or 6 depending upon conditions established for the study. Previous studies at Rockwell have resulted in WTGF values of 1.5 to 2.0 when the wing

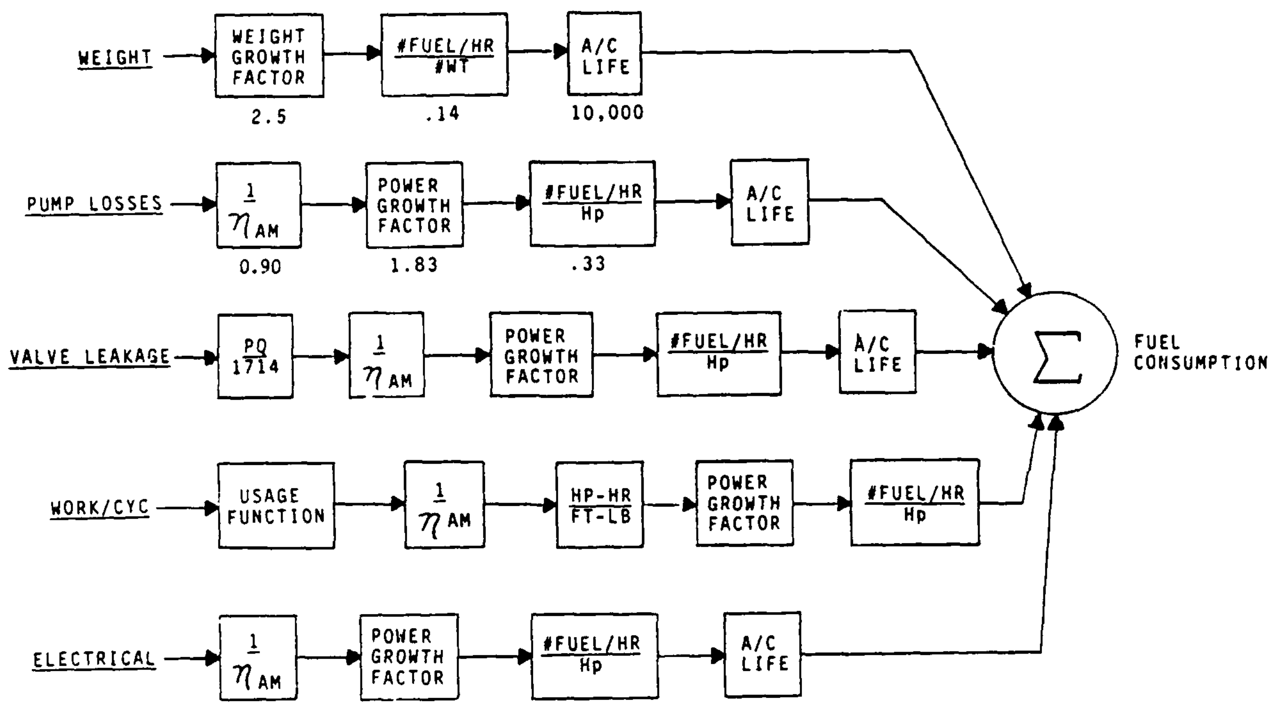


Figure 6. Fuel consumption calculations

area and engine parameters are held fixed, and values of 4.0 to 5.0 when wing area and engine parameters are permitted to vary. A weight growth factor of 2.5 was used in this study. This assumes a fixed engine and a variable wing area. Increased weight results in indirect fuel consumption through increased drag. The fuel consumption rate has three components:

1. Actual hydraulic equipment weight plus mounting hardware.

2. The increase in structural weight necessary to support the installed equipment plus the structural weight to carry the additional fuel.

3. The weight of the fuel. This component varies as a function of time.

Components of the weight growth factor are depicted in Figure 7 (a).

Power growth factor (PWGF) is defined as the factor by which the shaft extraction power must be multiplied to account for the indirect components associated with carrying the fuel to produce the power. The total fuel consumption rate per horsepower has three components.

1. Direct: shaft power extraction from the engine

2. Indirect-Structural: Shaft power requires a certain amount of fuel to supply the power over the mission time. This fuel has weight which must be supported by the airframe. Thus, the airframe weight increases by a certain fixed amount. This increment produces drag which results in a constant fuel rate increment.

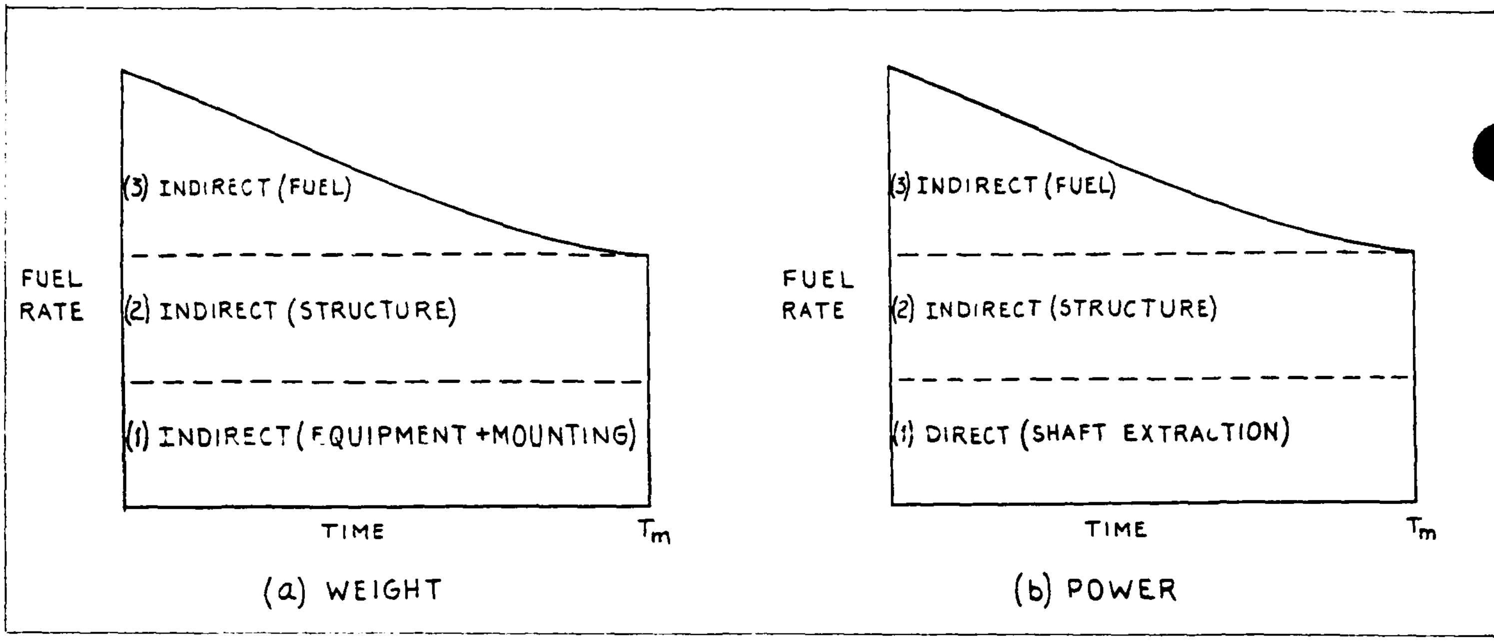


Figure 7. Growth factor components

WEIGHT

$$F = \overbrace{W_E}^{(1)} C_{\#} T_m + \overbrace{(W_E + W_{F_\phi}) C_s}^{(2)} C_{\#} T_m + \int_0^{T_m} \overbrace{W_F(t) C_{\#}}^{(3)} dt$$

$$W_F(t) = W_{F_\phi} e^{-C_{\#} t}$$

$$W_{F_\phi} \int_0^{T_m} e^{-C_{\#} t} \, d(C_{\#} t) = -W_{F_\phi} e^{-C_{\#} t} \Big|_0^{T_m}$$

$$= W_{F_\phi} (1 - e^{-C_{\#} T_m})$$

Therefore,

$$F \triangleq W_{F_\phi} = W_E C_{\#} T_m + (W_E + W_{F_\phi}) C_s C_{\#} T_m + W_{F_\phi} (1 - e^{-C_{\#} T_m})$$

$$W_{F_\phi} \left[1 - C_s C_{\#} T_m - (1 - e^{-C_{\#} T_m}) \right] = W_E C_{\#} T_m (1 + C_s)$$

$$W_{F_\phi} = \frac{W_E (1 + C_s) C_{\#} T_m}{e^{-C_{\#} T_m} - C_s C_{\#} T_m}$$

$$\triangleq W_E C_{\#} T_m W_{TGF}$$

POWER

$$F = \overbrace{P_e}^{(1)} C_p T_m + \overbrace{(W_{F_\phi} C_s) C_{\#}}^{(2)} T_m + \int_0^{T_m} \overbrace{W_F(t) C_{\#}}^{(3)} dt$$

However:

$$F \triangleq W_{F_\phi} = P_e C_p T_m + W_{F_\phi} C_s C_{\#} T_m + W_{F_\phi} (1 - e^{-C_{\#} T_m})$$

Solving for W_{F_ϕ},

$$W_{F_\phi} = \frac{P_e C_p T_m}{e^{-C_{\#} T_m} - C_s C_{\#} T_m}$$

$$\triangleq P_e C_p T_m P_{WGF}$$

Where,

P_e = POWER EXTRACTION, hp
C_p = FCR_{HP}, #/hr/hp
T_m = MISSON TIME, hr
W_{F_ϕ} = INITIAL FUEL WEIGHT, lb
C_s = STRUCTURAL WEIGHT COEFFICIENT
$C_{\#}$ = FCR_{lb}, #/hr/lb
$W_F(t)$ = INSTANTANEOUS FUEL WEIGHT FUNCTION, lb
W_E = EQUIPMENT + MOUNTING WEIGHT
F = FUEL WEIGHT, lb

Figure 8. Derivation of growth factors

3. Indirect-Fuel: Fuel has weight which results in drag on the aircraft and, in turn, consumes fuel. Since fuel weight decreases throughout the flight, this component decreases with time.

These components are depicted in Figure 7 (b).

Derivations of the growth factors are shown in Figure 8. A weight growth factor of 2.5 was chosen based on prior Rockwell studies. The power growth factor was then computed, using the equations in Figure 8, by adjusting the structural weight growth factor (C_S) to produce a weight growth factor of 2.5. The power growth factor was then calculated from this value of C_S. The ratio of WTGF to PWGF increases as WTGF increases, thus a larger WTGF would accentuate the importance of weight relative to extracted power in determining total fuel consumption. The study showed that weight dominated in total fuel consumption, thus a larger WTGF would make the dominance even more pronounced.

2.1.4.2 Work/Cycle. Most actuation tasks in an airplane involve positioning a load in accordance with a command. This is depicted in Figure 9 for a flight control actuator. The load magnitude is of particular importance since it sizes the actuator and determines the amount of energy required. The load can be described by inertia (J), an energy loss (B), and a load spring (K). Inertia is established by the physics of the control surface. The energy loss term consists of actuation friction and aerodynamic damping. The spring consists of the aerodynamic load. The aerodynamic terms vary with flight conditions. For purposes of this study, the friction term was assumed to be entirely viscous. The procedure for computing actuation energy consumption involves determining the energy consumed in one cycle of motion (work/cycle) and then multiplying this amount by the number of cycles

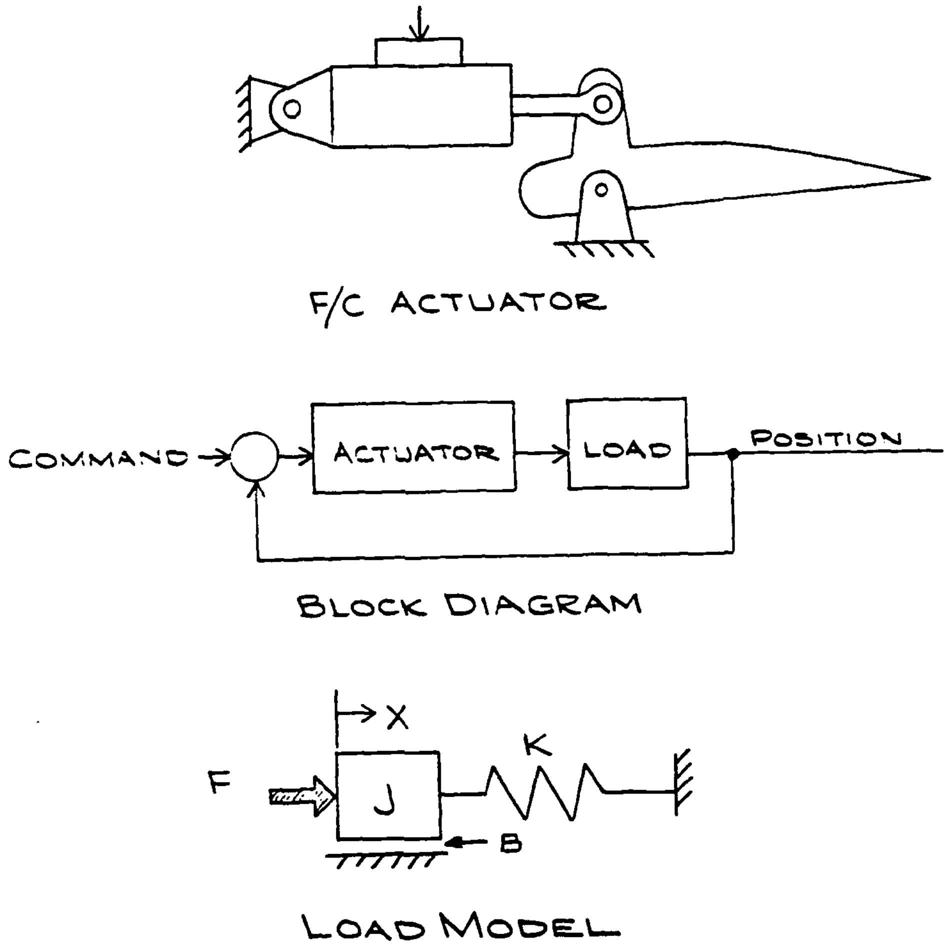


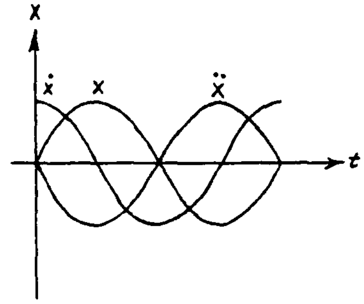
Figure 9. Actuation task

experienced during the life of the aircraft. The integral of power is energy or work. The output work performed by the actuator in moving the load through one cycle is derived in Figure 10 (a). It can be seen that actual output work is small and consists of only that required to overcome the losses ($\pi A^2 B \omega$). Unfortunately, the conventional actuator control element is very inefficient and does not recover stored energy. Most of the energy associated with actuation is consumed in controlling load position -- not in moving the load.

The work per cycle supplied by the hydraulic system is derived in Figure 10 (b) for a conventional balanced actuator. Work is equal to pressure times volume since $4D_m A$ is the volume of oil displaced in moving through one full cycle. The energy (work) consumed includes all inefficiencies of the distribution system, actuation control, and surface mechanism. These inefficiencies are accounted for in the design when actuator displacement (D_m) is selected. Efficiency is work out divided by work in and is given below:

$$\eta = \frac{\pi A B \omega}{4 D_m P}$$

$$x = A \sin \omega t$$
$$\dot{x} = A\omega \cos \omega t$$
$$\ddot{x} = -A\omega^2 \sin \omega t$$



$$T = P D_m = J\ddot{x} + B\dot{x} + Kx$$

$$Q = D_m \dot{x}$$

$$P_w = P\,Q = \left(\frac{J\ddot{x} + B\dot{x} + Kx}{D_m}\right) D_m \dot{x}$$
$$= \left(-JA\omega^2 \sin \omega t + BA\omega \cos \omega t + KA \sin \omega t\right) A\omega \cos \omega t$$

$$W = 4\int_0^{\frac{T}{4}} P_w \, dt$$
$$= 4A^2 \int_0^{\frac{\pi}{2}} \left[(K - J\omega^2) \sin \omega t + B\omega \cos \omega t \right] \cos \omega t \, d\omega t$$
$$= 4A^2 \left[\tfrac{1}{2} (K - J\omega^2) \sin^2 \omega t + \frac{B\omega}{2} (\omega t + \sin \omega t \cos \omega t) \right]_0^{\frac{\pi}{2}}$$
$$= 2A^2 \left[(K - J\omega^2) + B\omega \left(\frac{\pi}{2}\right) \right]$$

(a) Output

$$x = A \sin \omega t$$
$$T = P D_m$$
$$W = T x$$
$$W = 4\int_0^{\frac{\pi}{2}} P D_m x \, d\omega t$$
$$= 4 P D_m A \int_0^{\frac{\pi}{2}} \sin \omega t \, d\omega t$$
$$= 4 P D_m A$$

(b) Input (Hydraulic)

Where,

A = Amplitude
B = Damping
D_m = Actuator displacement
J = Inertia
K = Spring constant
P = Pressure
P_w = Power
Q = Flow
t = Time
T = Torque
W = Work
x = Displacement
ω = frequency

Figure 10. Work per cycle

2.1.5 Qualitative Assessment and Comparative Analysis

Factors relevant to the candidate energy saving techniques -- but difficult to quantify -- were qualitatively assessed. These factors are:

- Performance
- Reliability
- Maintainability
- Safety
- Life cycle cost
- Development risk

The assessment procedure employed the use of Subject Matter Experts (SME). A survey format was developed in which the SME's were asked to rate each factor for each candidate energy saving technique. The ratings were then averaged and multiplied by the energy savings estimate for the concept to produce a Figure of Merit (FOM) rating. The survey format and rating values are shown in Figure 11.

The committee of SME's consisted of members from the following disciplines:

- Air vehicle
- Control systems (2)
- Hydraulic systems (2)
- Reliability and Maintainability

CONCEPT RATING FORM

CANDIDATE CONCEPTS	ENERGY SAVING	R & M	LCC	DEVELOPMENT RISK	PERFORMANCE	SAFETY	RATING
A	(1)	(2)	(2)	(2)	(2)	(2)	
B							
C							
D							
ETC							

1 ENERGY SAVING ESTIMATE
2 QUALITATIVE RATINGS

RATING VALUES

RELIABILITY AND MAINTAINABILITY RATING

LEVELS	RATING
Significant Improvement	+2
Improvement	+1
No Effect	0
Degradation	−1
Significant Degradation	−2

LIFE-CYCLE COST RATING

LEVELS	RATING
Major Reduction	+2
Significant Reduction	+1
Negligible	0
Significant Increase	−1
Major Increase	−2

DEVELOPMENT RISK RATING

LEVELS	RATING
Already Developed	0
Slight Risk	−1
Major Risk	−2
Questionable Possibility	−3

PERFORMANCE RATING

LEVELS	RATING
Greatly Improved	+2
Improved	+1
No Change	0
Degraded	−1
Greatly Degraded	−2

SAFETY RATING

LEVELS	RATING
*Improvement	+1
No Change	0
**Degradation	−1
Unacceptable	−4

*Can Be Designed As Safe As You Want
**Size May Be Prohibitive or Contains Single-Point Failure

FIGURE 11. Comparative analysis

The SME's were chosen for their expertise and extensive background with similar systems or components and could be expected to reliably assess the qualitative factors for each candidate. The SME's were given the concept rating form without energy saving estimates, a description of each candidate, and instructed to assign a numerical rating for the qualitative factors. The rating values of the six SME's were averaged for each qualitative factor.

The figure of merit was calculated using the following formula:

$$FOM = [ES] \times [10 + 1/2\ (R_1+R_2+R_3+R_4+R_5)] \div 10$$

where, ES = Energy savings (M-lb fuel)

R_1 = Average R&M rating

R_2 = Average LCC rating

R_3 = Average development risk rating

R_4 = Average performance rating

R_5 = Average safety rating

The FOM is basically the energy savings scaled up or down by the qualitative factors. As an example, if the SME's evaluated a candidate concept and the concept received the best possible ratings in all areas, the FOM would be:

$$FOM = ES\ [10 + 1/2\ (2+2+0+2+1)] \div 10 = 1.35\ ES$$

The lowest FOM rating a concept could receive would be:

$$FOM = ES\ [10 + 1/2\ (-2-2-3-2-4)] \div 10 = 0.35\ ES$$

Provisions were included in the rating system to produce a very low rating for candidates that were considered a safety risk (-4) or an extreme development risk (-3), Figure 11. Thus, the "lowest qualitative" system would require 3 times the energy savings (ES) to have an FOM comparable to a neutral or no-risk system, and the "highest qualitative" system could have (100/1.35) % less energy savings and still have an FOM comparable to a "neutral" system.

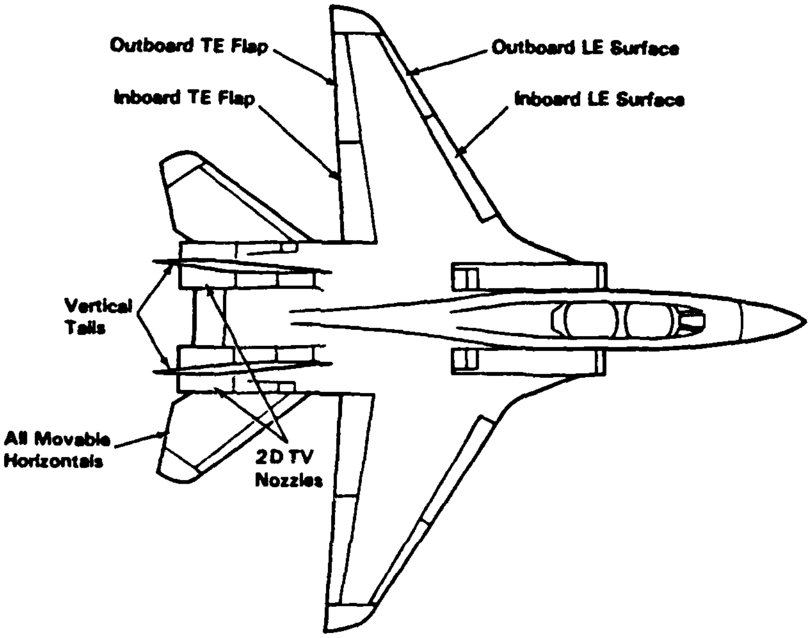
2.1.6 Industry Survey

A survey was conducted to gather information concerning energy saving approaches to aircraft hydraulic systems that are currently being pursued by the Aerospace industry. This was done to assure that all viable concepts were considered in this study. Most of the leading component and system manufacturers in the United States were contacted either by survey letter, by telephone, or by a personal visit to the supplier's plant. Some companies visited the Rockwell Columbus facility to discuss their current products and research efforts. The information provided was very helpful. Several companies were visited to solicit their participation in the Hardware Demonstration phase of this contract. A list of suppliers contacted and the survey questionnaire used are presented in Appendix A.

2.2 BASELINE VEHICLE

The baseline vehicle utilized is a hypothetical generic aircraft based upon data developed in the VFMX study effort conducted by Rockwell. A plan view of the aircraft and specifics are shown in Figure 12. Table 13 lists basic features of the vehicle. Mission requirements are discussed in Section 2.1.1. Basic aircraft systems are outlined in Table 14. The baseline hydraulic system is described in Section 2.3.

BASELINE VEHICLE



GENERAL DESCRIPTION	
Gross Weight	64,000 Lb
Span	50 Ft
Length	70 Ft
T/W	0.9
Fuel	21,000 Lb
Stores	6,400 Lb
Wing Area	675 Ft²

PERFORMANCE PARAMETERS

- $Mach_{Max}$ 1.8
- N_Z 6.5
- $N_{Z_{ULT}}$ 9.75
- V_{APP} 120 Knots
- Sink Rate 24 FPS

CONTROL EFFECTORS

Pitch
- Horizontals
- Inboard TE
- Vectored Thrust

Roll
- Outboard TE
- Horizontals
- Vectored Thrust

Yaw
- Rudders
- Vectored Thrust

NA 4677C

Figure 12. Baseline aircraft

TABLE 13. Baseline aircraft features

- o STUDY CRITICAL FEATURES
  - o MULTI-MISSION DESIGN
  - o ADVANCED 1990's ENGINES
  - o 2-D VECTORABLE, REVERSING NOZZLES
  - o DIGITAL INTEGRATED CONTROL SYSTEM
- o RELATED FEATURES
  - o RCS REDUCTION
  - o ADVANCED STRUCTURES/MATERIALS
  - o ADVANCED AVIONICS
  - o ADVANCED WEAPONS
  - o MODERN COCKPIT
  - o ADVANCED AIRCRAFT SUBSYSTEMS

TABLE 14. Baseline aircraft systems

- o HYDRAULIC SYSTEM
  - o 8000 PSI, 3 INDEPENDENT SYSTEMS
- o FLIGHT CONTROL
  - o 4 CHANNEL DIGITAL FBW
  - o INTEGRATES WITH PROPULSION AND FIRE CONTROL
  - o RELAXED STATIC STABILITY DESIGN
  - o REDUNDANCY FOR SURVIVABILITY
- o ELECTRICAL SYSTEM
  - o HVDC POWER
  - o APU BACKUP
- o ENVIRONMENTAL CONTROL SYSTEM
  - o CLOSED LOOP
  - o ELECTRICALLY DRIVEN

2.3 BASELINE HYDRAULIC SYSTEM

The system was configured to maximize combat survivability and operational readiness. Three 8000 psi hydraulic systems, designed to operate at temperatures from -40°F to +275°F using fluid per MIL-H-83282, are employed. Localized fluid temperatures in the engine area can possibly reach +300°F. The systems are depicted in Figure 13. Systems 1 and 3 are dedicated to primary flight controls. System 2 powers both flight controls and utility functions. The hydraulic power supply is shown in Figure 14. Each supply feeds two independent circuits immediately downstream of the pressure line filter.

Each system has two independent circuits monitored by reservoir-level-sensing (RLS) devices. Output from each pump flows through two RLS shutoff valves mounted downstream of the pressure line filter. Should a leak develop such that fluid in the reservoir drops below a normal operating level, valve "A" closes to isolate circuit "A" (see Figures 13 and 14). If the leak is not in circuit "A", the fluid level will continue to drop. At a preset lower level, circuit "A" valve reopens and valve "B" closes to isolate the leaking circuit. RLS operation is not affected by contamination, temperature variations, pressure fluctuations, or normal reservoir fluid level changes. RLS operates electrically and requires redundant electric power for sensing, control and monitoring. Check valves protect the return system by preventing back flow out of the reservoirs.

Reservoir-level-sensing significantly improves reliability, survivability, and maintainability. In addition to minimizing the effect of failures and combat damage, the shutoff feature reduces potential fire effects by limiting the quantity of fluid available to a leaking circuit. Maintenance costs associated with loss of fluid are reduced, since the pumps are not run dry, which would require pump replacement and system flushing.

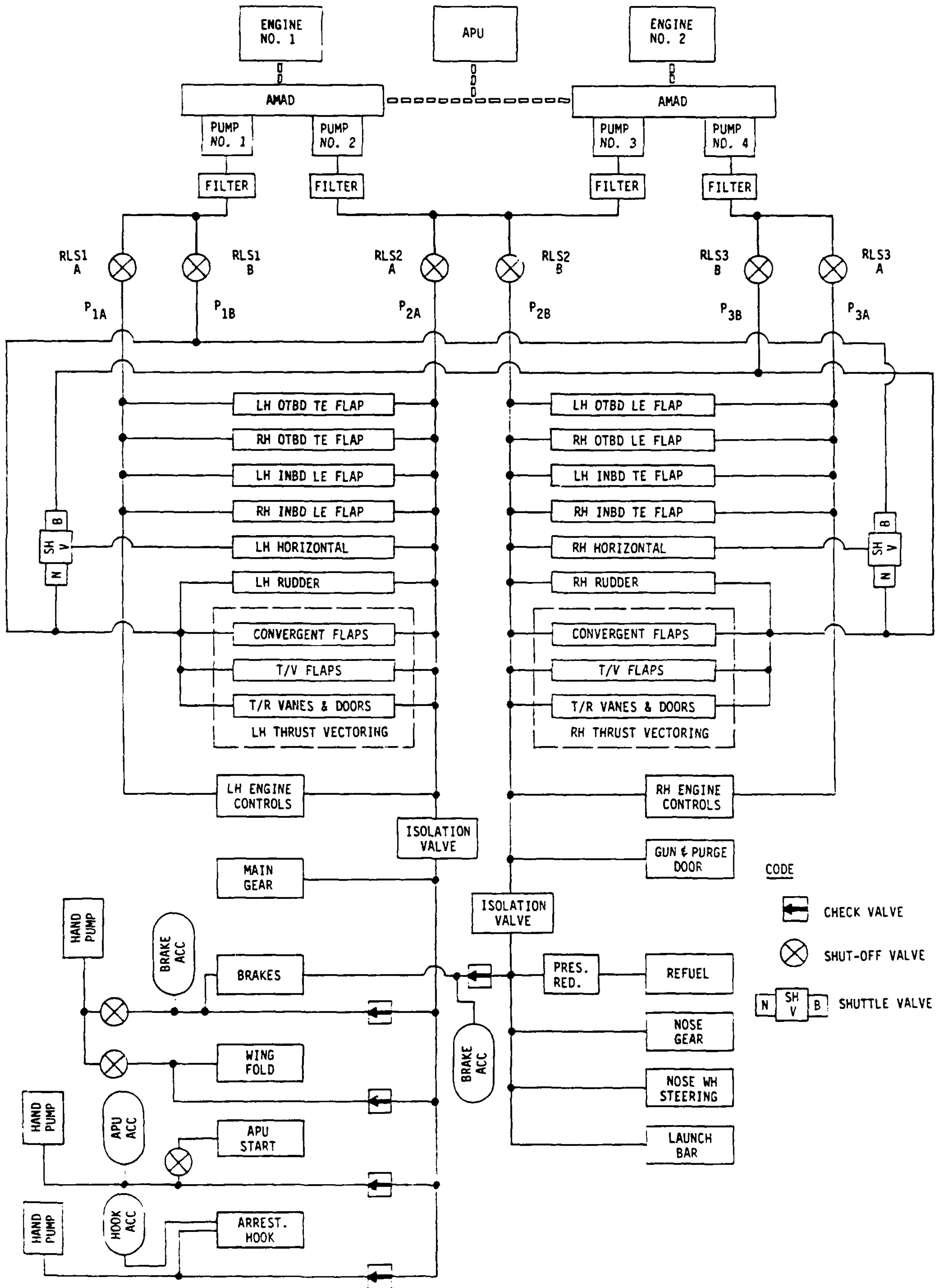


Figure 13. Baseline hydraulic system

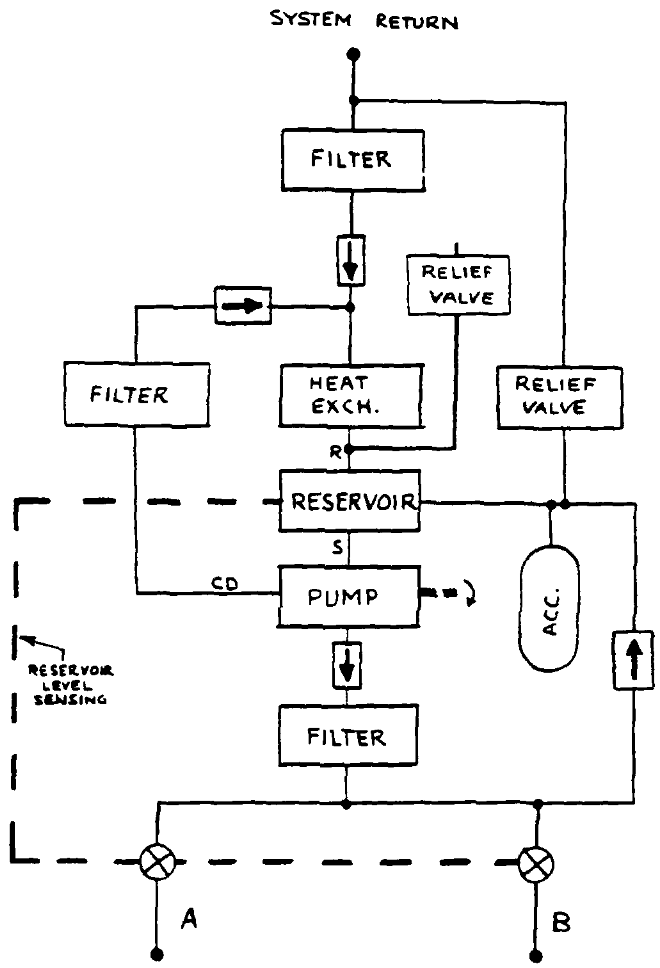


Figure 14. Hydraulic power supply

The reservoirs are located to provide positive pressure head to the pumps and are installed in different aircraft orientations to preclude simultaneous pump airlocks if air should enter a suction line from mis-serviced or improperly bled reservoirs.

Two identical airframe-mounted accessory drives (AMAD), each shaft-driven by an engine and a common APU, power the hydraulic pumps, Figure 13. The pumps are conventional in design and built to meet specification LHS-8810A, reference 14. Maximum flow is 40 gpm at 5700 rpm which is pump speed at rated engine speed. Fast pump response and quick-acting relief valves limit pressure overshoot to 8600 psi when flow demand stops suddenly. An auxiliary power unit (APU) supplies hydraulic power to the AMAD's for engine start and ground hydraulic power. This eliminates the need for hydraulic ground support equipment.

Control-by-wire (CBW) is utilized for all flight controls. Four electrical channels are employed for redundancy. Hydraulic power redundancy is provided by judicious use of six hydraulic circuits, three hydraulic power supplies, and shuttle valves. Dual tandem linear actuators drive the horizontal stabilizers, T/V flaps, and T/R vanes and doors. Dual rotary hingeline actuators power all other primary flight control surfaces. Hingeline actuation is necessary due to thin wing sections. Dual tandem actuators are used in the utility system to provide engine control after one failure. All other utility operations are powered by single unbalanced actuators except the gun drive and APU start which use hydraulic motors. Survivability is maximized by redundancy of the control effectors, as well as the hydraulic systems/actuation (see Figure 12). For example, roll control power is provided by both leading and trailing edge surfaces, differential horizontal stabilizers, and thrust vectoring.

Outboard TE flaps and the horizontal stabilizer operate both symmetrically and differentially. Inboard trailing edge flaps, leading edge flaps, and rudders operate symmetrically.

Shuttle valves provide an additional hydraulic power backup to the normal supply sources for the horizontal stabilizer actuators. The shuttle valves control both pressure and return flow paths. Normal supply pressure (N) positions the valve to port the normal supply and return to the actuator. With loss of normal pressure the valve switches (spring-biased) to a test position which blocks the normal supply ports and interconnects the two downstream ports. The backup (B) supply remains blocked. In this position, a test pressure is generated by a small spring-loaded accumulator (built into the valve) in the downstream circuit. Should downstream test pressure decay, the spool remains in the test position, preventing the loss of the backup system fluid. However, should the loss of test pressure be due to cavitation in the actuator circuit, a reset feature re-establishes the test pressure, and the spool shifts to the backup position. Normal spool position has priority so that, if the engine supplying the normal flow is shut down and restarted, the spool returns to the normal position.

Shuttle valves enhance survivability by providing three power supplies for pitch control. The integral test function in the valves permits maximum use of available hydraulic circuits as system backup sources.

System 1 is dedicated to primary and secondary flight control functions only. System 2 provides power for both flight control and utility functions. Two solenoid operated isolation valves separate all utility functions from flight control functions. The isolation valves are activated a few seconds after the landing gear doors are closed and locked, and de-activated when the landing gear is down. The isolation valves can be overridden by a switch in the cockpit. Normally, the APU accumulator is recharged after the left engine has been started.

An alternate hydraulic method of engine-starting is provided by using an accumulator to power the APU. The accumulator is charged through an isolation valve which is switched open by an aircraft OLEO switch or a cockpit override switch. The 8000 psi APU accumulator provides energy for two start attempts. A hand pump is included for self-sufficiency in ground operations. Hand pumps are also provided to replenish the brake accumulators, perform the wing fold operation and to raise the arresting hook for handling and maintenance.

Redundancy and survivability features incorporated in the baseline system are summarized in Table 15. Emergency actuation of specific functions are listed in Table 16. All other functions depend upon the reliaoility of the hydraulic system.

2.3.1 Hydraulic System Loads

Hydraulic system loads are listed in Tables 17 through 22. Tables 17 through 19 give actuation functions and their power requirements. Tables 20 through 22 delineate system design, and list the power supplied by the hydraulic system, flow for each load, total power, and total flow for each load group. For example, primary flight and thrust vectoring controls would extract 601 hp (131 gpm) if they were all operating at their respective design load conditions simultaneously. These loads are nearly evenly divided between F/C and T/V controls, and are fairly well balanced between the three hydraulic supplies, as shown by the total flows in Table 20.

TABLE 15. Redundancy and survivability features

1. Two engines power three independent hydraulic systems.
2. All flight control actuators are either 1) linear dual tandem, with rip-stop design and two stage rod seals or 2) dual rotary vane.
3. All flight control surfaces are dual and powered by three hydraulic supplies.
4. Each horizontal stabilizer actuator is supplied by three hydraulic sources.
5. An AMAD is utilized to remove pumps, generators, and other accessories from the high temperature environment of the engine bay, and to improve maintainability and minimize the fire hazard. Equipment is separated by intervening equipment and structure to increase combat damage survivability.
6. Each flight control function is supplied by four hydraulic distribution circuits.
7. Reservoir-level-sensing is used to disconnect leaking circuits to maintain fluid for the alternate circuit and to prevent operating pumps without fluid.
8. Two-fail-operative FBW system assures control after loss of two electronic channels.
9. An onboard APU provides engine start, emergency power, and ground power for self-sufficiency.
10. Flight controls automatically revert to dampers when hydraulic or electrical power is lost.
11. Line routings are widely separated to minimize the probability of combat damage (single hit) disabling the entire system.

TABLE 16. Emergency actuation

ITEM NO.	FUNCTION	METHOD	ACCUMULATOR SIZE
1.	LANDING GEAR	FREE FALL	-----
2.	BRAKES		
	A) DIFFERENTIAL (NORMAL)	HYD. ACCUM.	100 IN^3 @ 1000 psi
	B) SUM (EMERGENCY)	HYD. ACCUM.	50 IN^3 @ 1000 psi
3.	ARRESTING HOOK	HYD. ACCUM.	40 IN^3 @ ?
4.	HIGH LIFT	PRIMARY FLIGHT CONTROLS	-----
5.	APU START	HYD. ACCUM.	150 IN^3 @ 4000 psi
6.	CANOPY	AIR BOTTLE	-----
7.	GROUND STEERING	RUDDER & DIFF. BRAKES	-----

TABLE 17. Primary flight control loads

LOAD NO.	CONTROL SURFACE	ACT/ SURFACE	MAX HINGE MOMENT (LB-IN)	MAX RATE (DEG/SEC)	TRAVEL (DEG)	ACTUATOR DESIGN LOAD (DEG/SEC @ LB-IN)	HP PER SURFACE
1	OUTBOARD L.E.	2	135,000	20	+ 0° -30°	12 @ 54,000	3.43
2	INBOARD L.E.	2	150,000	20	+ 0° -30°	12 @ 60,000	3.81
3	OUTBOARD T.E.	2	135,000	50	±30°	30 @ 54,000	8.57
4	INBOARD T.E.	2	150,000	50	±30°	30 @ 60,000	9.52
5	RUDDER	1	100,000	55	±30°	33 @ 80,000	6.98
6	HORIZONTAL ALL MOVABLE	1	550,000	40	+11° -25°	24 @ 440,000	27.92

TABLE 18. Engine control loads

LOAD NO.	CONTROL SURFACE	ACT/ ENGINE	MAX FORCE LB	MAX RATE (IN/SEC)	TRAVEL	DESIGN LOAD IN/SEC @ LB	HP PER ENGINE
	THRUST VECTORING						
7	CONVERGENT FLAPS	4	13,125	5.3	6.0	3.2 @ 10,500	20.23
8	TV FLAPS (THRUST VECTORING)	4	10,500	11.7	15.0	7 @ 8,400	35.74
9	T/R VANES (THRUST REVERSING)	2	5,000	14.2	8.5	8.5 @ 4,000	10.3
10	T/R DOOR (THRUST REVERSING)	2	2,500	2.5	1.5	1.5 @ 2,000	.9
	VARIABLE GEOMETRY						
11	LPVG (LOW PRESS. VAR. GEOMETRY)	1	750	5.0	3.0	3.0 @ 600	.55
12	A94 (BYPASS)	3	1,250	3.3	2.0	2.0 @ 1,000	1.82
13	IGV (INLET GUIDE VANES)	2	813	5.8	3.5	3.5 @ 650	1.38
14	FVABI (FWD VAR. AREA BYPASS INJECTOR)	2	3,125	5.8	3.5	3.5 @ 2,500	5.27
15	HPVG (HIGH PRESS. VAR. GEOMETRY)	2	3,125	4.2	2.5	2.5 @ 2,500	3.79
16	AVABI (AFT VAR. AREA BYPASS INJECTOR)	1	2,250	4.5	2.7	2.7 @ 1,800	1.47

TABLE 19. Utility loads

LOAD NO.	FUNCTION	ACT/ AIRCRAFT	MAX FORCE (LB)	MAX RATE (IN/SEC)	TRAVEL (IN)	DESIGN LOAD IN/SEC @ LB	HP PER AIRCRAFT
17	LAUNCH BAR	1	1,000	6.67	8	4 @ 800	.49
18	NOSE WHEEL STEERING	1	4,688	1.65	7.42	1.0 @ 3,750	.56
	NOSE GEAR						
19	GEAR ACTUATOR	1	9,400	3.0	8.80	1.8 @ 7,520	2.05
20	FAIRING DOOR LOCK	1	920	4.8	1.20	2.9 @ 740	0.32
21	FAIRING DOOR	1	4,900	6.4	4.60	3.8 @ 3,920	2.28
22	GEAR LOCK CYL.	2	1,800	3.0	.75	.78 @ 1,440	0.79
	MAIN GEAR						
23	GEAR ACTUATOR	2	13,300	2.92	10.5	1.8 @ 10,640	5.65
24	FAIRING DOOR LOCK	2	920	5.0	1.25	3.0 @ 740	.67
25	FAIRING DOOR	2	7,200	7.7	7.50	4.6 @ 5,760	8.06
26	GEAR LOCK CYL.	4	1,800	1.3	.75	1.8 @ 1,440	.68
27	APU START	1					5.65
	REFUEL						
28	AFT DOOR & LOCK	1	2,000	3.5	2.1	2.1 @ 1,600	.51
29	CTR DOOR	1	920	2.33	1.4	1.4 @ 700	.16
30	CTR DOOR LOCK	1	920	5.2	1.3	3.1 @ 700	.35
31	PROBE ACT.	1	1,000	2.00	4.0	1.2 @ 800	.15
	GUN						
32	DRIVE MOTOR	1				29 GPM @ 2,600 PSI	44.0 (.5 SEC BURST)
33	PURGE DOOR	1	1,250	16.67	2.5	10. @ 1,000	1.52
34	MAIN BRAKES	2	20,000	.5	0.1	.30 @ 16,000	1.45
	WING FOLD						
35	LOCK PIN	4	2,350	15.	4.5	9. @ 1,880	10.25
36	WING FOLD ACT	2	13,290	.12	4.0	.07 @ 10,600	.23
37	ARRESTING HOOK	1	19,500	1.5	5.6	.9 @ 15,600	2.13

TABLE 20. Hydraulic system flow, F/C and T/V controls

LOAD NO.	ACT TYPE	DESIGN FACTOR (%)	NO LOADS PER A/C	TOTAL DESIGN LOAD (HP/AC)	TOTAL POWER SUPPLIED	FLOW (GPM) 1A	1B	2A	2B	3A	3B
1	D-RV	2.0	2	14.13	28.25	.00	.00	.00	2.21	2.21	.00
2	D-RV	2.0	2	15.78	22.50	2.46	.00	2.46	.00	.00	.00
3	D-RV	1.6	2	28.26	40.49	4.42	.00	4.42	.00	.00	.00
4	D-RV	1.6	2	31.40	44.99	.00	.00	.00	4.91	4.91	.00
5	D-RV	1.6	2	23.03	32.99	.00	1.80	1.80	1.80	.00	1.80
6	DT-UB	1.6	2	92.11	131.98	.00	7.20	7.20	7.20	.00	7.20
7	DT-UB	1.6	2	64.76	92.78	.00	5.06	5.06	5.06	.00	5.06
8	DT-UB	1.6	2	114.36	163.85	.00	8.94	8.94	8.94	.00	8.94
9	DT-UB	1.6	2	33.05	47.35	.00	2.58	2.58	2.58	.00	2.58
10	DT-UB	1.6	2	2.91	4.17	.00	.23	.23	.23	.00	.23
				419.7	601.3	6.9	25.8	32.7	32.9	7.1	25.8

TABLE 21. Hydraulic system flow, engine controls

LOAD NO.	ACT TYPE	DESIGN FACTOR (%)	NO LOADS PER A/C	TOTAL DESIGN LOAD (HP/AC)	TOTAL POWER SUPPLIED	FLOW (GPM) 1A	1B	2A	2B	3A	3B
11	DT-UB	2.0	2	1.09	1.56	.09	.00	.09	.09	.09	.00
12	DT-UB	2.0	2	3.60	5.16	.28	.00	.28	.28	.28	.00
13	DT-UB	2.0	2	2.74	3.93	.21	.00	.21	.21	.21	.00
14	DT-UB	2.0	2	10.55	15.11	.82	.00	.82	.82	.82	.00
15	DT-UB	2.0	2	7.64	10.94	.60	.00	.60	.60	.60	.00
16	DT-UB	2.0	2	2.05	4.22	.23	.00	.23	.23	.23	.00
				28.6	40.9	2.2	.0	2.2	2.2	2.2	.0

TABLE 22. Hydraulic system flow, utility functions

LOAD NO.	ACT TYPE	DESIGN FACTOR (%)	NO LOADS PER A/C	TOTAL DESIGN LOAD (HP/AC)	TOTAL POWER SUPPLIED	FLOW (GPM) 1A	1B	2A	2B	3A	3B
17	S-UB	1.0	1	.49	.69	.00	.00	.00	.15	.00	.00
18	S-UB	1.3	1	.56	.81	.00	.00	.00	.18	.00	.00
19	S-UB	1.3	1	2.05	2.94	.00	.00	.00	.64	.00	.00
20	S-UB	1.0	1	.32	.46	.00	.00	.00	.10	.00	.00
21	S-UB	1.0	1	2.28	3.27	.00	.00	.00	.71	.00	.00
22	S-UB	1.0	1	.79	1.13	.00	.00	.00	.25	.00	.00
23	S-UB	1.0	1	5.65	8.09	.00	.00	1.77	.00	.00	.00
24	S-UB	1.0	1	.67	.96	.00	.00	.21	.00	.00	.00
25	S-UB	1.0	1	8.06	11.55	.00	.00	2.52	.00	.00	.00
26	S-UB	1.0	1	.68	.88	.00	.00	.21	.00	.00	.00
27	MOTOR	-1.0	1	5.65	6.48	.00	.00	1.41	.00	.00	.00
28	S-UB	1.0	1	.51	.73	.00	.00	.00	.16	.00	.00
29	S-UB	1.0	1	.16	.22	.00	.00	.00	.05	.00	.00
30	S-UB	1.0	1	.35	.50	.00	.00	.00	.11	.00	.00
31	S-UB	1.0	1	.15	.21	.00	.00	.00	.05	.00	.00
32	MOTOR	-1.0	1	44.00	58.43	.00	.00	.00	11.01	.00	.00
33	S-UB	1.0	1	1.52	2.17	.00	.00	.00	.47	.00	.00
34	S-UB	1.0	1	1.45	2.08	.00	.00	.46	.00	.00	.00
35	S-UB	1.0	1	10.25	14.69	.00	.00	3.21	.00	.00	.00
36	S-UB	1.0	1	.23	.33	.00	.00	.07	.00	.00	.00
37	S-UB	1.0	1	2.13	3.05	.00	.00	.67	.00	.00	.00
				87.9	111.0	.0	.0	10.5	13.9	.0	.0

Redundancy requirements necessitate oversize actuators to provide performance capability after system failures. All control functions are provided by multiple control elements, some to a greater extent than others. For example, roll control is provided by the outboard trailing edge flaps (ailerons), differential horizontal stabilizer, and thrust vectoring. Full roll control power is provided after a single hydraulic system failure by the stabilizer and T/V controls and 80 percent from the outboard trailing edge flaps. The outboard trailing edge flaps are therefore sized to provide 160 percent of the required power when all systems are operational. The "design factor" shown in Table 20 accounts for the redundancy requirements accordingly.

The design load and the total power supplied were determined using the following relationships:

$$\begin{pmatrix}\text{Total}\\ \text{Design}\\ \text{Load}\end{pmatrix} = \begin{pmatrix}\text{Actuator}\\ \text{Design}\\ \text{Load}\end{pmatrix} \times \begin{pmatrix}\text{No. Actuators}\\ \text{Per}\\ \text{Load Group}\end{pmatrix} \times \begin{pmatrix}\text{No. Loads}\\ \text{Per}\\ \text{Aircraft}\end{pmatrix} \times \begin{pmatrix}\text{Design}\\ \text{Factor}\end{pmatrix} \times \begin{pmatrix}\text{Surface}\\ \text{Mechanical}\\ \text{Efficiency}\end{pmatrix}$$

$$\begin{pmatrix}\text{Total}\\ \text{Supplied}\\ \text{Power}\end{pmatrix} = \begin{pmatrix}\text{Total}\\ \text{Design}\\ \text{Load}\end{pmatrix} + \begin{pmatrix}\text{Distribution}\\ \text{System}\\ \text{Losses}\end{pmatrix} + \begin{pmatrix}\text{Control}\\ \text{Valve}\\ \text{Losses}\end{pmatrix}$$

One thousand psi drop in the supply lines and 1370 psi drop in the control valve were assumed in the calculations. To compute the power extracted from the engine, the efficiencies of the AMAD and hydraulic pumps must be included. Assuming 90% and 66%, respectively, for these elements, the total extracted power from the engine would be 1012 hp for the F/C and T/V controls. Fortunately, design load power is not demanded from all controls simultaneously.

2.3.2 Distribution System

The distribution system is shown schematically in Figure 15 for the primary controls and in Figure 16 for the engine and utility functions. Utility subsystems which consist of a number of actuation loads that occur concurrently, such as the landing gear, are shown in block form and treated as a single load for purposes of this study.

Hydraulic lines were sized using the criteria given in Table 23, and "EVEN" sizes for both pressure and return lines. The flow limit for each size tube is listed in Table 24 along with the length, which produces 1000 psi pressure drop, and the Reynolds number for the limit flow condition at +50°F.

Design data for the distribution system hydraulic lines are summarized in Table 25 for flight controls, and Table 26 for engine and utility systems. Combined, the 188 lines have the following totals:

- Length - 1363 feet
- Weight - 364 pounds
- Volume - 4050 cubic inches
- Fluid Volume - 10.11 gallons

Tables 25 and 26 show the pressure drop for the supply lines at the design load (Rate @ Load) flow value. Flows in branch feed lines (lines which supply fluid for more than one load) were determined using the following algorithm:

$$Q_{FEED} = Q_{MX} + \frac{1}{2}\left(\sum_{n} Q_i - Q_{MX}\right)$$

where, Q_{MX} = the maximum load flow of the loads
Q_i = load flow for the i^{th} load
Q_{Feed} = flow used to size the feed line

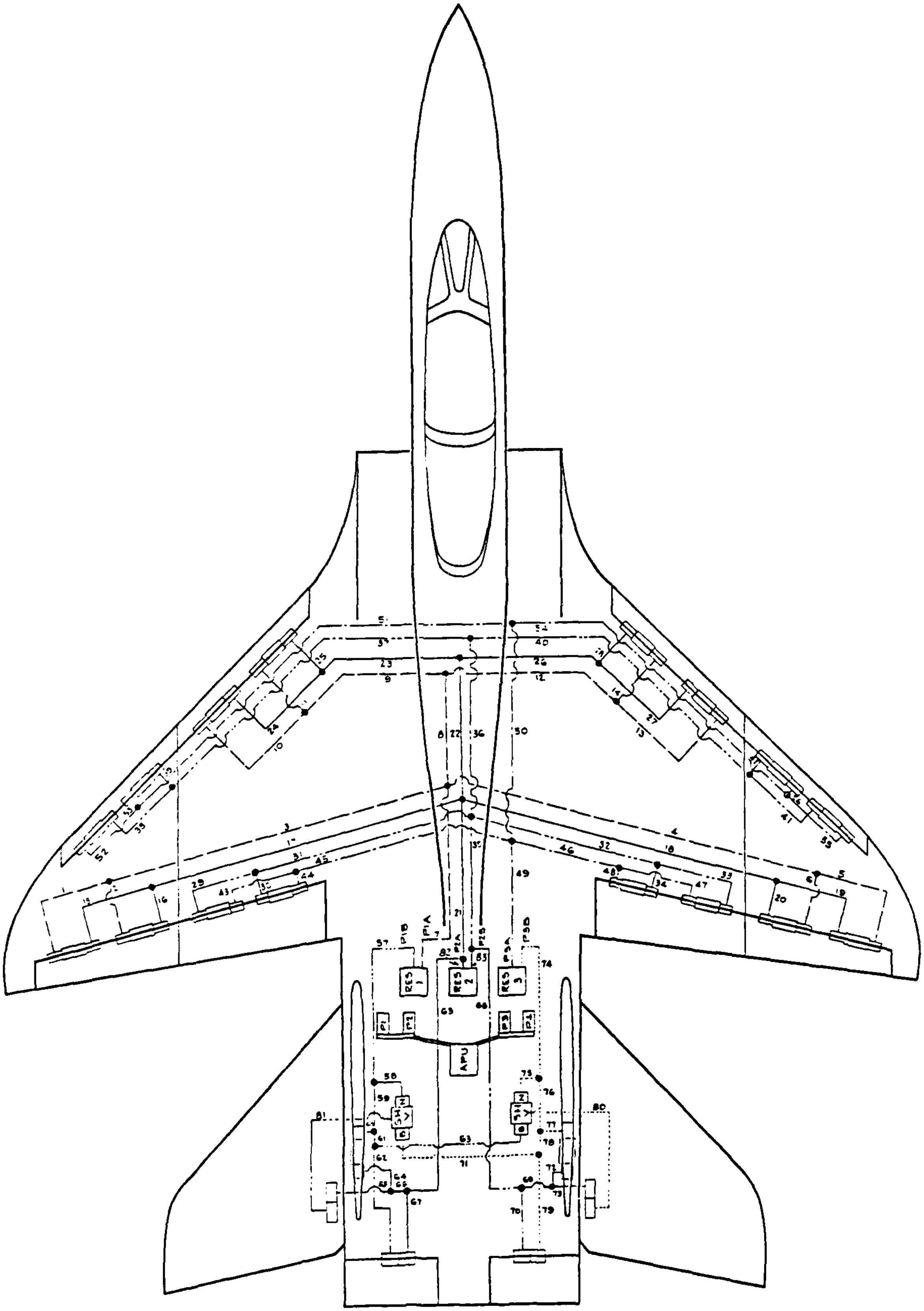


Figure 15. Primary controls

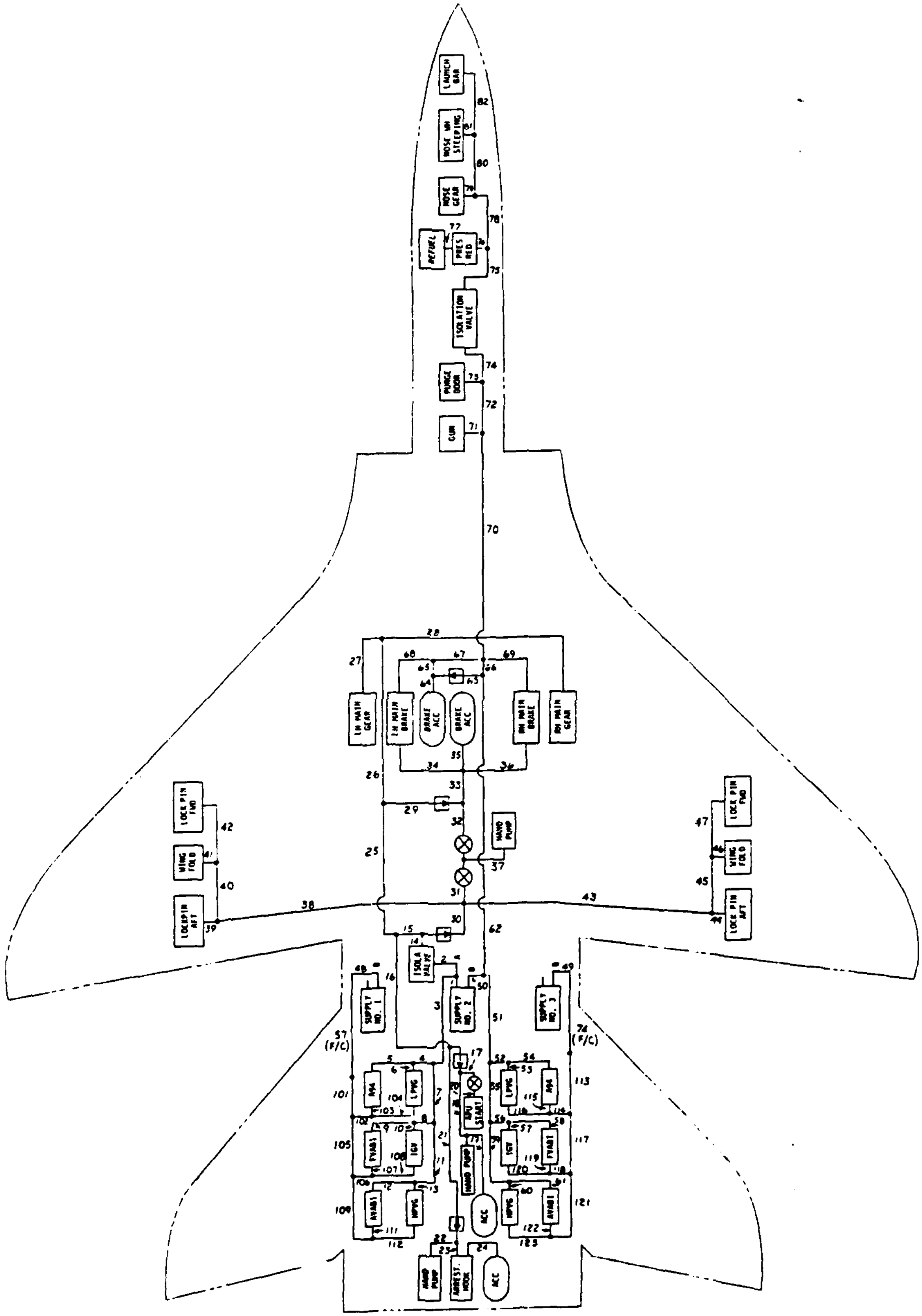


Figure 16. Engine controls and utility system

TABLE 23. Baseline sizing procedure

Lines
1. Maximum fluid velocity: 25 ft/sec
2. Fluid temperature: +50°F
3. 1000 psi maximum pressure drop at the design load flow
4. 5333 psi maximum pressure drop (2/3 x 8000 psi) at maximum no-load flow
Valve/Actuator
1. Fluid temperature: +50°F
2. Size actuator for maximum hinge moment at 6850 psi differential pressure
3. Size valve for the greater of:
A. Maximum no load rate at 2283 psi (1/3 x 6850 psi).
B. Rate at the design load point.

TABLE 24. "EVEN" size line flow limits

Flow at 25 ft/sec velocity

Size	Flow, gpm	Length, ft	RN
4	2.70	23.5	315.1
6	6.87	59.7	502.6
8	12.84	111.6	687.2
10	20.09	174.8	859.7
12	28.97	251.9	1032.3
14	39.46	343.2	1204.8
16	51.35	446.6	1374.3

TABLE 25. Baseline distribution lines, F/C and T/V systems

NO	LENGTH FT	FLOW GPM	SIZE-P	SIZE-R	DELT-P P	DELT-P R	DELT-P SUM	WT	INSTL VOL	REYN	
1	6.20	1.11	6	4	26.44	42.94	69.38	1.25	13.70	94.	432.
2	.59	1.11	4	4	12.74	4.08	16.82	.07	.81	142.	432.
3	24.26	2.22	6	4	207.69	337.86	545.55	4.90	53.59	189.	864.
4	47.45	2.22	6	6	406.22	130.53	536.75	13.14	144.41	189.	576.
5	6.20	1.11	6	4	26.44	42.94	69.38	1.25	13.70	94.	432.
6	.59	1.11	4	4	12.74	4.08	16.82	.07	.81	142.	432.
7	9.20	6.87	6	6	247.63	107.96	355.60	2.23	26.06	584.	1783.
8	19.50	2.45	6	6	184.38	59.27	243.65	4.73	55.23	208.	636.
9	2.07	1.24	4	4	49.04	16.03	65.07	.26	2.85	158.	483.
10	5.78	.62	4	4	69.58	22.31	91.88	.73	7.94	79.	241.
11	.59	.62	4	4	7.10	2.28	9.38	.07	.81	79.	241.
12	28.05	1.24	6	4	133.68	217.16	350.84	5.66	61.96	105.	483.
13	6.60	.62	6	4	15.69	25.47	41.16	1.33	14.58	53.	241.
14	.59	.62	4	4	7.10	2.28	9.38	.07	.81	79.	241.
15	6.20	1.11	4	4	133.84	42.94	176.78	.78	8.52	142.	432.
16	.59	1.11	4	4	12.74	4.08	16.82	.07	.81	142.	432.
17	18.15	2.22	6	4	155.38	252.77	408.15	3.66	40.09	189.	864.
18	39.60	2.22	6	6	339.02	108.94	447.95	10.96	120.52	189.	576.
19	6.20	1.11	6	6	26.44	8.48	34.92	1.72	18.87	94.	288.
20	.59	1.11	4	4	12.74	4.08	16.82	.07	.81	142.	432.
21	15.60	5.65	8	6	111.04	129.87	240.91	5.32	61.59	362.	1467.
22	19.50	2.45	6	6	184.38	59.27	243.65	4.73	55.23	208.	636.
23	1.25	1.24	4	4	30.16	9.68	39.84	.16	1.72	158.	483.
24	5.78	.62	4	4	69.58	22.31	91.88	.73	7.94	79.	241.
25	.59	.62	4	4	7.10	2.28	9.38	.07	.81	79.	241.
26	28.05	1.24	6	6	133.68	42.89	176.57	7.77	85.37	105.	322.
27	6.60	.62	6	6	15.69	5.03	20.72	1.83	20.09	53.	161.
28	.59	.62	6	6	1.40	.45	1.85	.16	1.80	53.	161.
29	6.20	1.24	4	4	149.58	48.00	197.58	.78	8.52	158.	483.
30	.59	1.24	4	4	14.23	4.57	18.80	.07	.81	158.	483.
31	9.08	2.45	6	4	85.86	139.71	225.57	1.83	20.06	208.	954.
32	28.05	2.45	6	6	265.22	85.25	350.48	7.77	85.37	208.	636.
33	6.20	1.24	4	4	149.58	48.00	197.58	.78	8.52	158.	483.
34	.59	1.24	4	4	14.23	4.57	18.80	.07	.81	158.	483.
35	15.60	6.02	8	6	118.46	145.17	263.64	5.32	61.59	386.	1563.
36	19.50	2.22	6	6	166.94	53.64	220.58	4.73	55.23	189.	576.
37	21.45	1.11	6	6	91.47	29.35	120.81	5.94	65.28	94.	288.
38	6.20	.55	4	4	66.19	21.22	87.41	.78	8.52	70.	214.
39	.60	.55	6	4	1.27	2.05	3.32	.12	1.33	47.	214.
40	39.60	1.11	6	6	168.86	54.18	223.04	10.96	120.52	94.	288.
41	6.20	.55	4	4	66.19	21.22	87.41	.78	8.52	70.	214.
42	.60	.55	4	4	6.41	2.05	8.46	.08	.82	70.	214.
43	6.45	1.24	4	4	155.61	49.94	205.55	.82	8.87	158.	483.
44	.59	1.24	4	4	14.23	4.57	18.80	.07	.81	158.	483.
45	5.78	2.22	6	4	40.48	80.50	120.98	1.17	12.77	189.	864.
46	23.93	10.20	8	8	312.26	147.77	460.04	12.12	130.97	654.	1996.
47	6.20	1.24	6	6	29.55	9.48	39.03	1.72	18.87	105.	322.
48	.59	1.24	4	4	14.23	4.57	18.80	.07	.81	158.	483.
49	17.55	27.47	12	10	131.53	219.30	350.83	15.15	171.45	1180.	4313.
50	19.50	10.87	8	8	251.10	117.73	368.84	8.57	98.75	646.	1970.
51	27.23	1.11	6	6	116.11	37.25	153.37	7.54	82.87	94.	288.
52	6.20	.55	6	4	13.07	21.22	34.29	1.25	13.70	47.	214.
53	.59	.55	6	6	1.24	.40	1.64	.16	1.80	47.	143.
54	42.90	1.11	6	6	182.93	58.69	241.62	11.88	130.56	94.	288.
55	6.20	.55	6	6	13.07	4.19	17.27	1.72	18.87	47.	143.
56	.59	.55	6	6	1.24	.40	1.64	.16	1.80	47.	143.
57	8.78	21.11	10	8	101.91	104.88	206.79	5.01	57.22	1086.	4130.
58	.98	7.20	6	6	27.68	12.48	40.16	.24	2.78	612.	1869.
59	3.90	21.11	10	8	45.27	86.56	131.83	2.22	25.42	1086.	4130.
60	.99	1.44	4	4	27.76	8.91	36.67	.11	1.26	184.	561.
61	.96	20.40	10	8	10.74	20.06	30.81	.55	6.26	1050.	3992.
62	12.68	16.80	10	8	115.50	188.32	303.81	7.23	82.64	864.	3287.
63	11.70	21.11	10	8	135.80	259.60	395.40	6.67	76.25	1086.	4130.
64	3.90	1.44	4	4	109.34	35.10	144.44	.43	4.95	184.	561.
65	5.85	7.20	6	6	165.21	74.55	239.76	1.42	16.57	612.	1869.
66	1.95	7.91	6	6	60.65	29.31	89.96	.47	5.52	673.	2053.
67	5.85	16.80	10	6	53.28	330.28	383.56	2.76	31.60	864.	4361.
68	11.70	21.11	10	8	135.80	259.69	395.49	6.67	76.25	1086.	4130.
69	1.95	7.91	8	6	19.58	29.31	48.89	.67	7.70	507.	2053.
70	5.85	16.80	10	8	53.28	86.89	140.17	3.34	38.13	864.	3287.
71	11.70	7.20	8	8	106.69	39.17	145.86	5.14	59.25	462.	1409.
72	1.95	1.44	4	4	54.67	17.55	72.22	.21	2.47	184.	561.
73	5.85	7.20	8	6	53.34	74.55	127.89	2.00	23.10	462.	1869.
74	8.85	21.11	10	8	102.72	106.43	209.16	5.05	57.68	1086.	4130.
75	.98	7.20	10	8	3.71	3.28	6.99	.56	6.39	370.	1409.
76	3.90	21.11	10	8	45.27	86.56	131.83	2.22	25.42	1086.	4130.
77	.98	1.44	4	4	27.48	8.82	36.30	.11	1.24	184.	561.
78	.98	20.40	10	8	10.97	20.48	31.45	.56	6.39	1050.	3992.
79	12.68	16.80	10	8	115.50	188.32	303.81	7.23	82.64	864.	3287.
80	11.70	7.20	10	8	44.24	39.17	83.40	6.67	76.25	370.	1409.
81	11.70	7.20	6	6	330.42	149.09	479.51	2.84	33.14	612.	1869.
82	1.80	24.51	10	8	24.52	51.96	76.49	1.83	11.73	1261.	4796.
83	1.50	25.71	10	8	23.24	47.10	70.34	.86	9.78	1323.	5031.
	-----							-----	-----		
	783.33							248.51	2788.22		

TABLE 26. Baseline distribution lines, engine and utility systems

NO	LENGTH FT	FLOW GPM	SIZE-P	SIZE-R	DELT-P P	DELT-P R	DELT-P SUM	WT	INSTL VOL	REYN	
1	1.05	3.20	6	6	13.00	4.18	17.18	.29	3.20	272.	831.
2	3.38	3.20	6	6	41.85	13.47	55.32	.94	10.29	272.	831.
3	10.35	1.53	4	4	308.41	99.01	407.42	1.31	14.23	195.	596.
4	1.80	.33	4	4	11.52	3.69	15.21	.23	2.47	42.	128.
5	4.95	.28	4	4	26.88	8.61	35.49	.63	6.80	36.	109.
6	1.05	.09	4	4	1.83	.59	2.42	.13	1.44	11.	35.
7	5.25	1.34	4	4	136.92	43.94	180.87	.66	7.22	171.	522.
8	1.80	.72	4	4	25.17	8.07	33.24	.23	2.47	92.	280.
9	4.95	.82	4	4	78.86	25.29	104.15	.63	6.80	105.	319.
10	1.05	.21	4	4	4.27	1.37	5.64	.13	1.44	27.	82.
11	7.35	.72	4	4	102.78	32.96	135.74	.93	10.10	92.	280.
12	4.95	.23	4	4	22.07	7.07	29.15	.63	6.80	29.	90.
13	1.05	.60	4	4	12.23	3.92	16.15	.13	1.44	77.	234.
14	1.50	3.20	6	6	18.57	5.98	24.55	.42	4.57	272.	831.
15	2.40	3.20	6	6	29.72	9.56	39.28	.66	7.30	272.	831.
16	15.60	1.41	6	4	84.59	137.45	222.03	3.15	34.46	120.	549.
17	1.50	1.41	4	4	41.18	13.22	54.39	.19	2.06	180.	549.
18	5.70	1.41	4	4	156.47	50.22	206.69	.72	7.83	180.	549.
19	6.75	1.41	4	4	185.29	59.47	244.76	.85	9.28	180.	549.
20	3.60	1.41	4	4	98.82	31.72	130.54	.46	4.95	180.	549.
21	18.15	.67	4	4	236.14	75.71	311.85	2.30	24.95	85.	261.
22	4.20	.10	4	4	8.14	2.61	10.75	.53	5.77	13.	39.
23	1.65	.67	4	4	21.47	6.88	28.35	.21	2.27	85.	261.
24	.15	.67	4	4	1.95	.63	2.58	.02	.21	85.	261.
25	13.05	1.60	6	6	80.35	25.80	106.14	3.61	39.72	136.	415.
26	15.00	1.60	6	4	92.35	150.11	242.46	3.03	33.13	136.	623.
27	7.35	.80	4	4	114.23	36.63	150.86	.93	10.10	102.	311.
28	20.85	.80	4	4	324.04	103.92	427.96	2.64	28.66	102.	311.
29	6.75	.10	4	4	13.08	4.19	17.27	.85	9.28	13.	39.
30	8.25	3.21	6	6	102.47	32.97	135.45	2.28	25.11	273.	833.
31	3.60	.10	4	4	6.98	2.24	9.21	.46	4.95	13.	39.
32	5.25	.10	4	4	10.17	3.26	13.43	.66	7.22	13.	39.
33	2.55	.30	4	4	14.84	4.75	19.59	.32	3.50	38.	117.
34	7.80	.23	4	4	34.78	11.15	45.93	.99	10.72	29.	90.
35	2.55	.23	4	4	11.37	3.64	15.02	.32	3.50	29.	90.
36	7.80	.23	4	4	34.78	11.15	45.93	.99	10.72	29.	90.
37	4.65	.10	4	4	9.01	2.89	11.90	.59	6.39	13.	39.
38	27.00	1.60	6	4	166.24	272.19	438.43	4.75	55.37	136.	623.
39	1.20	.80	4	4	18.65	5.98	24.63	.13	1.52	102.	311.
40	6.15	.80	4	4	95.58	32.65	128.23	.67	7.80	102.	311.
41	1.20	.25	4	4	5.82	1.86	7.68	.13	1.52	32.	97.
42	7.95	.80	4	4	123.56	39.62	163.18	.87	10.09	102.	311.
43	26.40	1.60	6	4	162.54	264.19	426.73	4.65	54.14	136.	623.
44	1.20	.80	4	4	18.65	5.98	24.63	.13	1.52	102.	311.
45	6.15	.80	4	4	95.58	32.65	128.23	.67	7.80	102.	311.
46	1.20	.25	4	4	5.82	1.86	7.68	.13	1.52	32.	97.
47	7.95	.80	4	4	123.56	39.62	163.18	.87	10.09	102.	311.
48	5.85	22.60	8	8	202.94	146.41	349.34	2.96	32.02	1449.	4422.
49	5.85	22.60	8	8	202.94	146.41	349.34	2.96	32.02	1449.	4422.
50	2.85	11.80	10	8	17.93	22.74	40.67	1.88	23.16	607.	2309.
51	7.80	1.53	4	4	232.43	74.62	307.04	.99	10.72	195.	596.
52	1.80	.33	4	4	11.52	3.69	15.21	.23	2.47	42.	128.
53	1.05	.09	4	4	1.83	.59	2.42	.13	1.44	11.	35.
54	4.95	.28	4	4	26.88	8.61	35.49	.63	6.80	36.	109.
55	5.25	1.34	4	4	136.92	43.94	180.87	.66	7.22	171.	522.
56	1.80	.72	4	4	25.17	8.07	33.24	.23	2.47	92.	280.
57	1.05	.21	4	4	4.27	1.37	5.64	.13	1.44	27.	82.
58	4.95	.82	4	4	78.86	25.29	104.15	.63	6.80	105.	319.
59	7.35	.72	4	4	102.78	32.96	135.74	.93	10.10	92.	280.
60	1.05	.60	4	4	12.23	3.92	16.15	.13	1.44	77.	234.
61	4.95	.23	4	4	22.07	7.07	29.15	.63	6.80	29.	90.
62	27.00	11.00	10	8	157.96	190.40	348.37	17.85	191.01	566.	2152.
63	4.20	.10	4	4	8.14	2.61	10.75	.53	5.77	13.	39.
64	1.50	.46	4	4	13.30	4.29	17.60	.19	2.06	59.	179.
65	1.50	.46	4	4	13.30	4.29	17.60	.19	2.06	59.	179.
66	1.50	11.00	8	8	21.17	10.58	31.74	.76	8.21	705.	2152.
67	4.65	.23	4	4	20.74	6.64	27.38	.59	6.39	29.	90.
68	6.30	.23	4	4	28.10	9.00	37.10	.80	8.66	29.	90.
69	6.30	.23	4	4	28.10	9.00	37.10	.80	8.66	29.	90.
70	20.25	11.00	8	8	285.73	142.80	428.53	10.25	110.83	705.	2152.
71	1.50	11.00	8	6	21.17	40.24	61.40	.59	6.39	705.	2855.
72	5.25	.64	4	4	65.24	20.92	86.16	.66	7.22	82.	249.
73	1.50	.47	4	4	13.68	4.39	18.07	.19	2.06	60.	183.
74	2.10	.64	4	4	26.10	8.37	34.46	.27	2.89	82.	249.
75	10.50	.64	4	4	130.48	41.83	172.31	1.33	14.43	82.	249.
76	1.31	.20	4	4	5.08	1.63	6.71	.17	1.80	26.	78.
77	1.31	.20	4	4	5.08	1.63	6.71	.17	1.80	26.	78.
78	5.70	.64	4	4	70.83	22.71	93.54	.72	7.83	82.	249.
79	1.05	.64	4	4	13.05	4.18	17.23	.13	1.44	82.	249.
80	6.75	.18	4	4	23.55	7.55	31.10	.85	9.28	23.	70.
81	1.31	.18	4	4	4.57	1.46	6.04	.17	1.80	23.	70.
82	7.35	.15	4	4	21.37	6.85	28.22	.93	10.10	19.	58.
101	16.50	1.53	6	6	97.12	31.18	128.30	4.57	50.22	130.	397.
102	1.80	.33	4	4	11.52	3.69	15.21	.23	2.47	42.	128.
103	1.05	.28	4	4	5.70	1.83	7.53	.13	1.44	36.	109.
104	4.95	.09	4	4	8.63	2.77	11.40	.63	6.80	11.	35.
105	5.70	1.34	4	4	148.66	47.71	196.37	.72	7.83	171.	522.
106	1.80	.92	4	4	32.18	10.32	42.51	.23	2.47	117.	358.
107	1.05	.82	4	4	16.73	5.36	22.09	.13	1.44	105.	319.
108	4.95	.21	4	4	20.15	6.46	26.61	.63	6.80	27.	82.
109	7.35	.72	4	4	102.78	32.96	135.74	.93	10.10	92.	280.
111	1.05	.23	4	4	4.60	1.50	6.10	.13	1.44	29.	90.
112	4.95	.60	4	4	57.66	18.49	76.14	.63	6.80	77.	234.
113	16.50	1.53	6	4	97.12	157.84	254.96	3.33	36.45	130.	596.
114	1.80	.33	4	4	11.52	3.69	15.21	.23	2.47	42.	128.
115	1.05	.28	4	4	5.70	1.83	7.53	.13	1.44	36.	109.
116	4.95	.09	4	4	8.63	2.77	11.40	.63	6.80	11.	35.
117	5.70	1.34	6	4	29.37	47.71	77.08	1.15	12.59	114.	522.
118	1.80	.92	4	4	32.18	10.32	42.51	.23	2.47	117.	358.
119	1.05	.82	4	4	16.73	5.36	22.09	.13	1.44	105.	319.
120	4.95	.21	4	4	20.15	6.46	26.61	.63	6.80	27.	82.
121	7.35	.72	4	4	102.78	32.96	135.74	.93	10.10	92.	280.
122	1.05	.23	4	4	4.60	1.50	6.10	.13	1.44	29.	90.
123	4.95	.60	4	4	57.66	18.49	76.14	.63	6.80	77.	234.
	579.71							115.42	1262.02		

In other words, the feed lines were sized to supply the maximum load flow plus half of the remaining load flows. In situations where the demand of various loads are known to occur simultaneously, they were added together and treated as a single load.

The distribution networks shown schematically in Figures 17 through 21 were used to establish line pressure drops. The total pressure drop was computed for each load by summing the drop in each leg of the circuit. Table 27 lists the supply line pressure drop for each load.

2.3.3 Actuation System

Actuation system load requirements are listed in Tables 17 through 19. Redundancy requirements and actuator type are given in Table 28. Weight and volume information for the baseline system actuators is presented in Table 29 and is based on trade study data in Appendix B and Reference 1. Total weight of all actuators, including actuator control valves is 1926.5 lb; total installation volume is 11,158 in^3.

2.3.4 Power Supply System

The power supply, by definition, includes all equipment between the aircraft fuel tanks and the main pump discharge ports, and consists of the following:

1) Engines
2) APU's
3) AMAD's
4) Pumps
5) Accumulators
6) Reservoirs
7) Heat Exchangers
8) Filters
9) Miscellaneous lines and valves

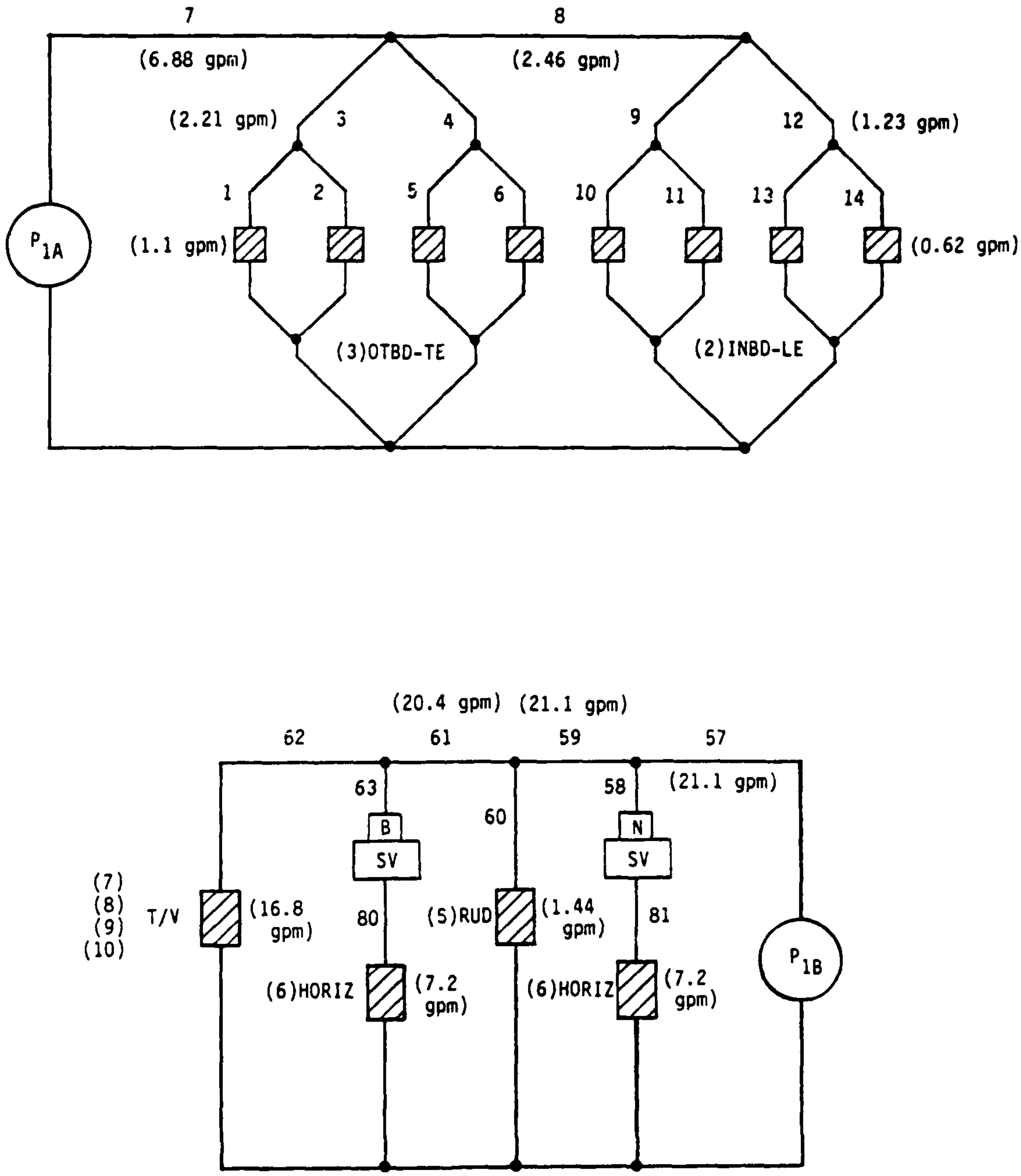


Figure 17. Distribution system, flight control system No. 1

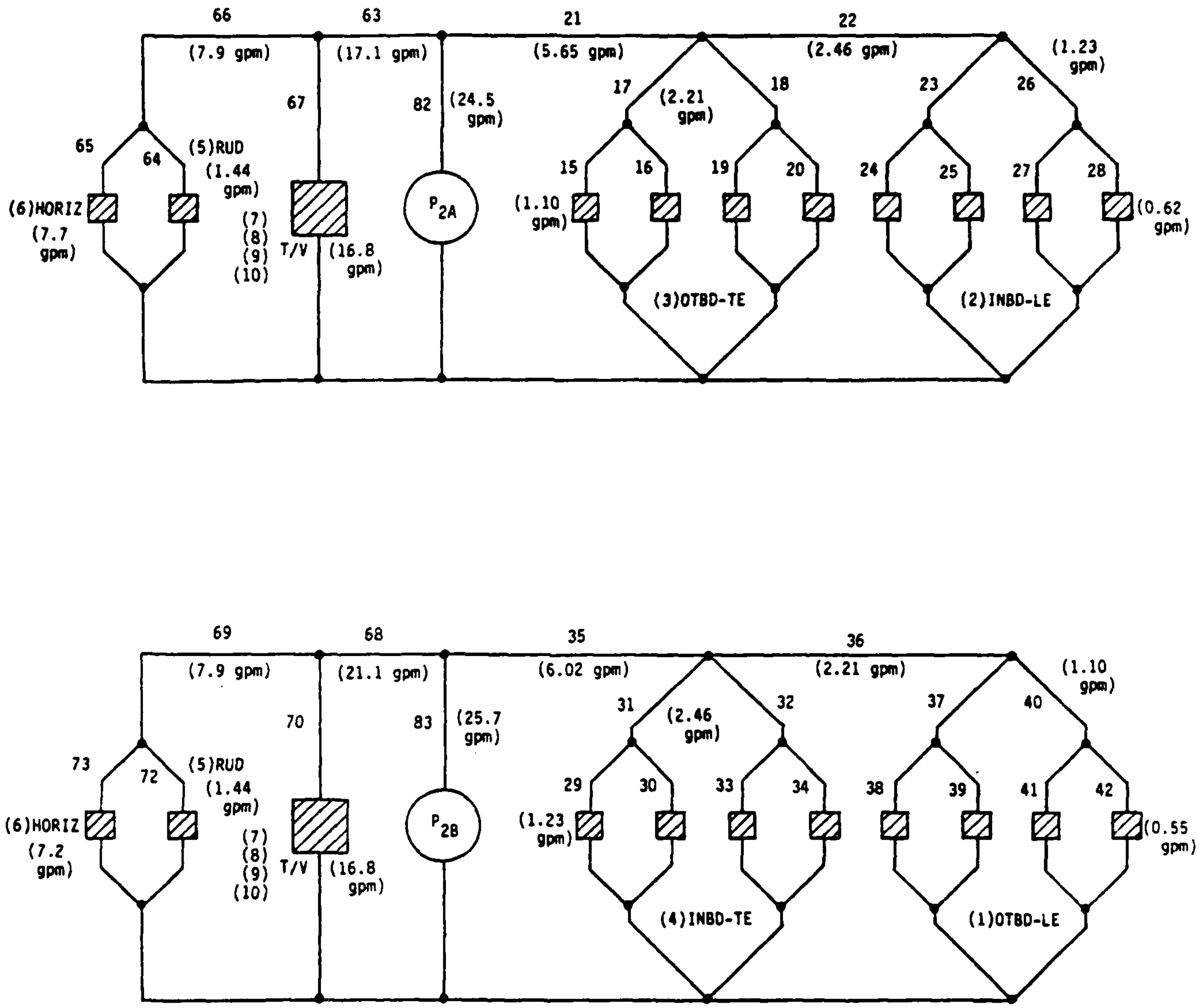


Figure 18. <u>Distribution system, flight control system No. 2</u>

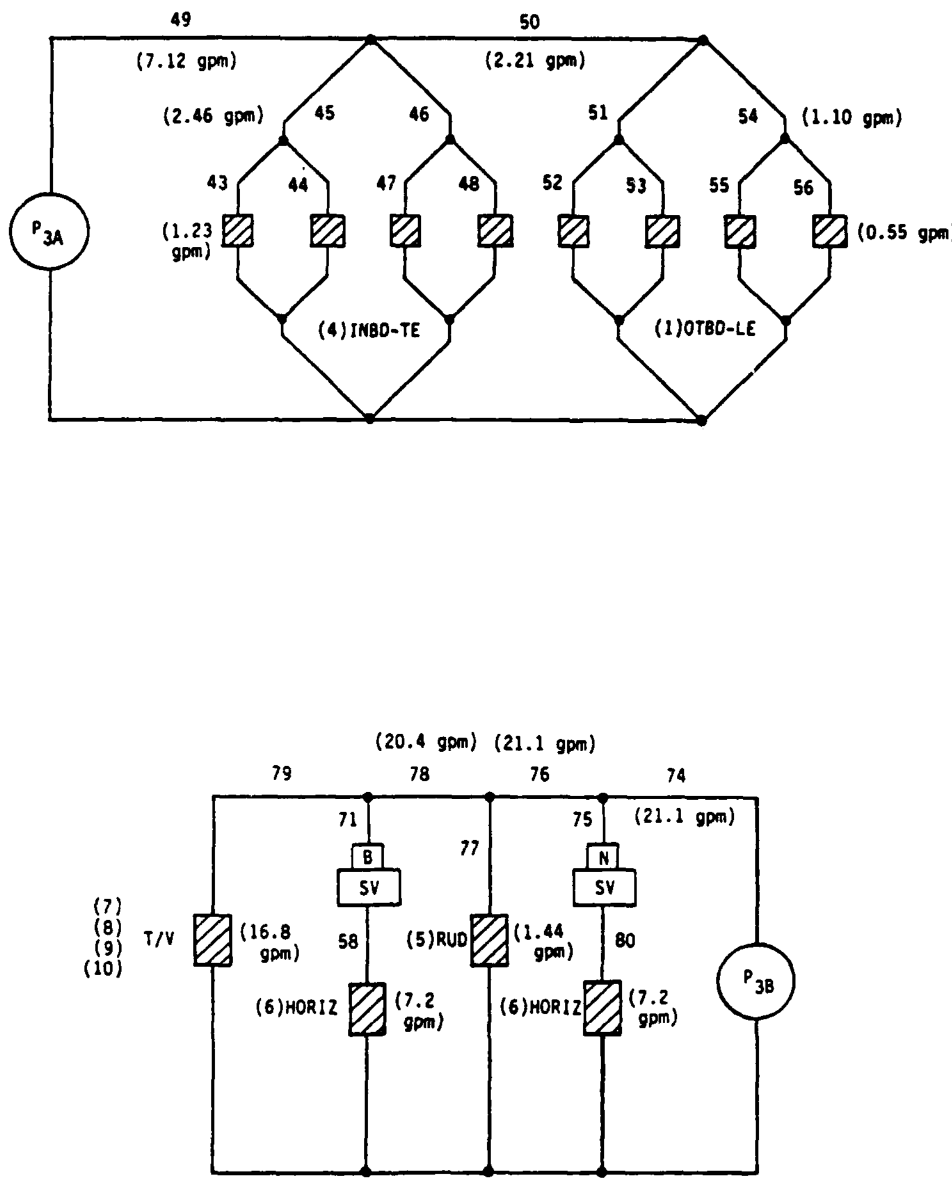


Figure 19. Distribution system, flight control system No. 3

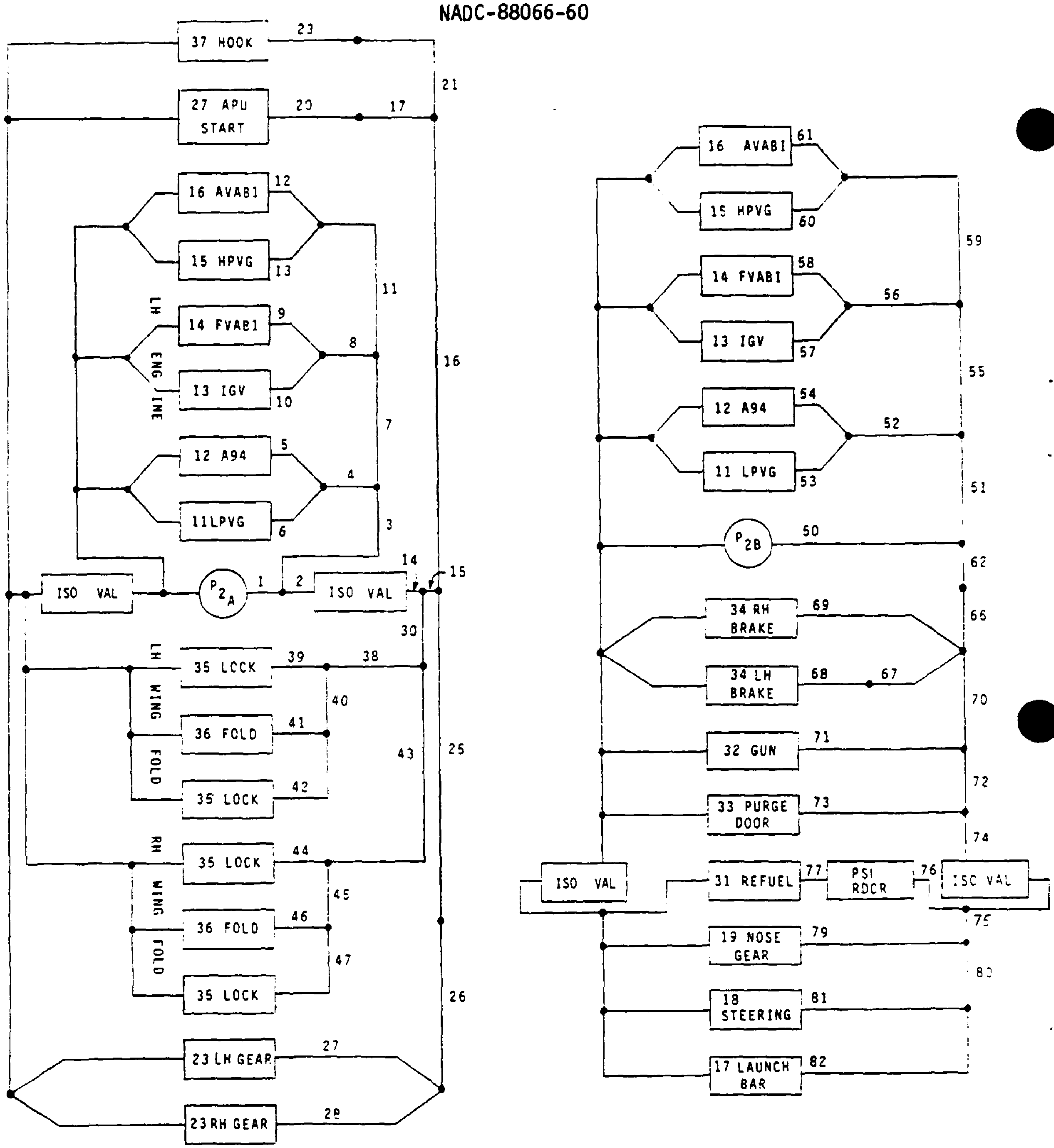


Figure 20. Distribution system, engine controls and utility functions, system No. 2

NOTE: Since the duty cycle of the gun and downstream loads do not occur simultaneously - the feed input data will ignore 50, 62, 66 and 70. Pressure drop at 1.35 GPM thru these larger lines is very small.

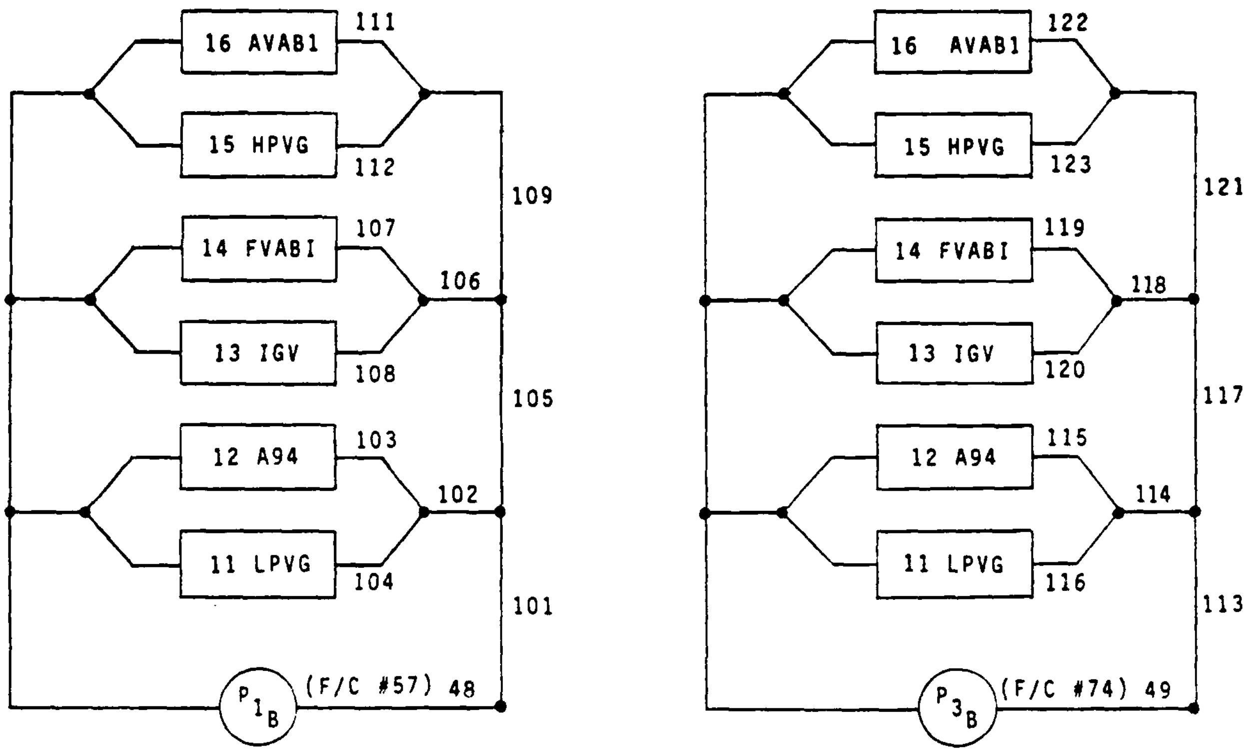


Figure 21. Distribution system, engine controls, systems No. 1 and No. 3

TABLE 27. Distribution line pressure drop

F/C AND T/V SYSTEMS

LOAD	FEED LINES										TOTAL LINE DROP
1	7	3	1	0	0	0	0	0	0	0	970.5
1	7	3	2	0	0	0	0	0	0	0	918.0
1	7	4	5	0	0	0	0	0	0	0	961.7
1	7	4	6	0	0	0	0	0	0	0	989.2
1	7	8	9	10	0	0	0	0	0	0	757.1
1	7	8	9	11	0	0	0	0	0	0	674.6
1	7	8	12	13	0	0	0	0	0	0	991.2
1	7	8	12	14	0	0	0	0	0	0	959.5
2	57	58	81	0	0	0	0	0	0	0	816.5
2	57	59	60	0	0	0	0	0	0	0	465.3
2	57	59	61	63	80	0	0	0	0	0	938.3
2	57	59	61	62	0	0	0	0	0	0	763.2
3	82	21	17	15	0	0	0	0	0	0	902.3
3	82	21	17	16	0	0	0	0	0	0	742.4
3	82	21	18	19	0	0	0	0	0	0	800.3
3	82	21	18	20	0	0	0	0	0	0	702.2
3	82	21	22	23	24	0	0	0	0	0	802.0
3	82	21	22	23	25	0	0	0	0	0	610.3
3	82	21	22	26	27	0	0	0	0	0	758.3
3	82	21	22	26	28	0	0	0	0	0	739.5
4	82	63	67	0	0	0	0	0	0	0	855.5
4	82	63	66	65	0	0	0	0	0	0	881.7
4	82	63	66	64	0	0	0	0	0	0	706.4
5	83	35	31	29	0	0	0	0	0	0	757.1
5	83	35	31	30	0	0	0	0	0	0	570.3
5	83	35	32	33	0	0	0	0	0	0	882.8
5	83	35	32	34	0	0	0	0	0	0	783.3
5	83	35	36	37	38	0	0	0	0	0	762.8
5	83	35	36	37	39	0	0	0	0	0	678.7
5	83	35	36	40	41	0	0	0	0	0	865.0
5	83	35	36	40	42	0	0	0	0	0	706.1
6	83	68	70	0	0	0	0	0	0	0	606.0
6	83	68	69	72	0	0	0	0	0	0	586.9
6	83	68	69	73	0	0	0	0	0	0	642.6
7	49	45	43	0	0	0	0	0	0	0	686.4
7	49	45	44	0	0	0	0	0	0	0	499.6
7	49	46	47	0	0	0	0	0	0	0	848.9
7	49	46	48	0	0	0	0	0	0	0	829.7
7	49	50	51	52	0	0	0	0	0	0	937.3
7	49	50	51	53	0	0	0	0	0	0	874.7
7	49	50	54	55	0	0	0	0	0	0	970.6
7	49	50	54	56	0	0	0	0	0	0	962.9
8	74	75	80	0	0	0	0	0	0	0	389.5
8	74	76	77	0	0	0	0	0	0	0	467.3
8	74	76	78	71	58	0	0	0	0	0	648.5
8	74	76	78	79	0	0	0	0	0	0	768.2

ENGINE AND UTILITY SYSTEMS

LOAD	FEED LINES										TOTAL LINE DROP
1	48	101	102	103	0	0	0	0	0	0	500.4
1	48	101	102	104	0	0	0	0	0	0	504.3
1	48	101	105	106	107	0	0	0	0	0	738.6
1	48	101	105	106	108	0	0	0	0	0	743.1
1	48	101	105	109	111	0	0	0	0	0	815.0
1	48	101	105	109	112	0	0	0	0	0	885.0
2	49	113	114	115	0	0	0	0	0	0	627.0
2	49	113	114	116	0	0	0	0	0	0	630.9
2	49	113	117	118	119	0	0	0	0	0	746.0
2	49	113	117	118	120	0	0	0	0	0	750.5
2	49	113	117	121	122	0	0	0	0	0	823.3
2	49	113	117	121	123	0	0	0	0	0	893.3
3	1	2	14	15	25	26	27	0	0	0	635.8
3	1	2	14	15	25	26	28	0	0	0	912.9
3	1	2	14	30	38	39	0	0	0	0	693.6
3	1	2	14	30	38	40	41	0	0	0	802.0
3	1	2	14	30	38	40	42	0	0	0	958.3
3	1	2	14	30	43	44	0	0	0	0	683.0
3	1	2	14	30	43	45	46	0	0	0	793.1
3	1	2	14	30	43	45	47	0	0	0	940.6
4	1	2	14	15	16	17	20	0	0	0	543.3
4	1	2	14	15	16	21	23	0	0	0	600.6
4	1	3	4	5	0	0	0	0	0	0	475.3
4	1	3	4	6	0	0	0	0	0	0	442.2
4	1	3	7	8	0	0	0	0	0	0	742.0
4	1	3	7	8	10	0	0	0	0	0	644.4
4	1	3	7	11	12	0	0	0	0	0	770.4
4	1	3	7	11	13	0	0	0	0	0	757.4
5	50	62	66	69	0	0	0	0	0	0	457.0
5	50	62	66	67	68	0	0	0	0	0	485.3
5	50	62	66	70	71	0	0	0	0	0	910.7
5	72	73	0	0	0	0	0	0	0	0	104.2
5	72	74	75	76	77	0	0	0	0	0	306.3
5	72	74	75	78	79	0	0	0	0	0	493.7
5	72	74	75	78	80	81	0	0	0	0	423.6
5	72	74	75	78	80	82	0	0	0	0	445.8
6	50	51	52	53	0	0	0	0	0	0	365.3
6	50	51	52	54	0	0	0	0	0	0	308.4
6	50	51	55	56	57	0	0	0	0	0	587.5
6	50	51	55	56	58	0	0	0	0	0	666.0
6	50	51	55	59	60	0	0	0	0	0	680.5
6	50	51	55	59	61	0	0	0	0	0	603.5

TABLE 28. Redundancy requirements

SURFACE ACTUATOR	ELECTRICAL CHANNELS	HYDRAULIC SUPPLIES	SINGLE HYD SUPPLY FAILURE	DUAL HYD SUPPLY FAILURE	DUAL ELECTRICAL FAILURE*
L.E. Flap	4	2	1/2 Load Capability	Up Lock	No Change
T.E. Flap	2	2	1/2 Load Capability	Trail Damped	Trail Damped
Rudder	2	2	No Change	Trail Damped	Trail Damped
Horizontal	4	2 With Backup	1/2 Load Capability**	1/2 Load Capability	No Change

*No change for single electrical failure. If all electric off, horizontal goes to neutral lock, L.E. flaps go to up position, all other actuators go to trail damped.

**No change if single failure is in backed up system.

TABLE 29. Baseline system actuator weight and volume

LOAD	DM	STROKE	SWEPT VOL	WT/ACT	TOTAL WT	TOTAL INSTL VOL	TOTAL FLUID VOL
1	20.32	-30.00	10.64	43.74	174.94	665.07	42.55
2	22.58	-30.00	11.82	45.53	182.13	696.14	47.28
3	16.25	-60.00	17.02	46.13	184.54	692.48	68.08
4	18.06	-60.00	18.91	49.77	199.08	743.74	75.64
5	24.08	-60.00	25.21	57.71	115.43	446.32	50.43
6	132.44	-36.00	83.21	52.74	105.48	789.73	166.42
7	3.07	6.00	18.39	22.93	183.43	1398.07	147.15
8	2.45	15.00	36.79	32.17	257.34	1966.02	294.31
9	1.17	8.50	9.93	17.92	71.67	538.71	39.71
10	.58	1.50	.88	4.34	4.34	34.76	3.50
11	.22	3.00	.66	6.02	12.04	85.91	1.31
12	.36	2.00	.73	3.98	3.98	31.82	4.38
13	.24	3.50	.83	6.48	25.93	186.60	3.32
14	.91	3.50	3.19	10.53	42.11	314.75	12.77
15	.91	2.50	2.28	9.25	36.98	274.24	9.12
16	.66	2.70	1.77	8.42	16.84	124.05	3.55
17	.15	8.00	1.17	6.51	6.51	49.44	1.17
18	.68	7.42	5.08	12.12	12.12	94.33	5.08
19	1.37	8.80	12.08	10.03	10.03	141.39	12.08
20	.13	1.20	.16	3.19	3.19	22.89	.16
21	.72	4.60	3.29	10.15	10.15	78.27	3.29
22	.26	.75	.20	3.40	6.80	49.20	.39
23	1.94	10.50	20.39	23.04	46.08	362.65	40.77
24	.13	1.25	.17	3.23	6.46	46.44	.34
25	1.05	7.50	7.88	15.46	30.93	238.87	15.77
26	.26	.75	.20	3.39	13.54	98.04	.79
27	.00	-360.00	.01	21.08	21.08	73.74	.01
28	.29	2.10	.61	5.07	5.07	37.93	.61
29	.13	1.40	.19	3.33	3.33	24.10	.19
30	.13	1.30	.17	3.27	3.27	23.55	.17
31	.15	4.00	.58	4.05	4.05	37.05	.58
32	.01	-360.00	.04	35.18	35.18	150.58	.04
33	.18	2.50	.46	4.62	4.62	34.14	.46
34	2.92	.10	.29	3.90	7.80	57.07	.58
35	.34	4.50	1.54	7.43	29.73	225.99	6.18
36	1.94	4.00	7.76	14.62	29.24	228.81	15.52
37	2.05	5.60	15.04	20.50	20.50	161.12	15.04
			-----		-----	-----	-----
			318.9		1926.5	11158.2	1088.8

Items 1 through 6 are discussed in the following subparagraphs of this section. Interest in this equipment, for purposes of this study, is limited to their weight and efficiency as they affect calculation of energy savings and power losses.

2.3.4.1 Engines. The engine selected for the baseline aircraft is in the 30,000 lb class and incorporates thrust vectoring in the exhaust nozzles. Thrust vectoring actuation is considered part of the primary flight controls and is included in the baseline hydraulic system. Variable geometry engine controls are powered by the aircraft hydraulic system.

The efficiency with which the engine converts fuel into shaft horsepower and thrust is dependent upon flight conditions. Specific fuel consumption data is presented in Section 2.1.2 along with aerodynamics for the vehicle. Fuel consumption in terms of shaft horsepower extraction and equipment weight is shown in Tables 8 and 10 for various flight modes. This data is the basis for calculating fuel consumption (total energy usage).

2.3.4.2 APU's. The APU supplies power for emergency flight conditions and ground operations. Emergency flight conditions occur so infrequently they have negligible effect on total energy usage. Ground operations were not considered as part of the study mission. Therefore, examination of the APU was limited to sizing the accumulator for starting (see section 2.3.4.5).

2.3.4.3 AMAD's. The AMAD's have an overall efficiency typical of gear boxes, and for purposes of this study, was assumed to be a constant value of 90%. AMAD's are normally designed to specific aircraft requirements, thus their weight and volume is dependant upon the power extracted. Data presented in Appendix B was used to estimate weight for computing total energy consumption.

2.3.4.4 Pumps. The baseline pumps are a conventional pressure compensated, variable delivery, axial piston design. Displacement is changed by angular movement of a trunnion mounted yoke on which the piston shoes slide as the cylinder barrel rotates. The barrel houses nine pistons. Yoke position is based on discharge pressure which drives the yoke piston through a servo controlled valve (compensator). Internal leakage is used to lubricate and cool the pump. Kidney shaped holes in a port plate time the entrance/exit of fluid into/from the nine piston chambers in the barrel.

Pump dynamic response will meet the transient and stability time requirements specified in LHS-8810A, reference 14. Pump discharge pressure will reach 90% of steady-state full flow pressure within 0.050 sec following a flow demand. Fast response limits discharge overshoot to 8600 psi when flow demand drops suddenly from full to quiescent flow. The pumps were sized in accordance with LHS-8810A; 1.7 in^3/rev was selected.

2.3.4.5 Accumulators. Control functions in the baseline aircraft that require emergency back-up and the method of emergency actuation are listed in Table 16. Four hydraulic accumulators are used. Accumulator size data is given in Table 16. Metal bellows type accumulators are employed to reduce maintenance costs.

2.3.4.6 Reservoirs. The total fluid volume and differential fluid volume for the three systems in the baseline vehicle are shown in Table 30. Differential volume is the maximum unbalanced volume resulting from full displacement of accumulators, unbalanced actuators and bootstrap reservoirs. The reservoirs were sized in accordance with MIL-R-5520C and the need for a minimum fluid rest volume. Systems No. 1 and No. 3 have a total fluid volume of 4.6 gal; system No. 2 has a total volume of 15.5 gal. The total baseline system fluid volume is 24.7 gal. Estimated reservoir weight and installation volumes shown in Table 31 were based upon the trade data in Appendix B.

TABLE 30. Fluid volume

COMPONENT	SYS. NO. 1		SYS. NO. 2		SYS. NO. 3	
	Total (in^3)	Diff. (in^3)	Total (in^3)	Diff (in^3)	Total (in^3)	Diff. (in^3)
Actuators	283	91.5	605	338	285	91.5
Accumulators	0	0	222	222	0	0
Lines & Fittings	405	-	1526	-	405	-
Reservoir	332.9	6.2	1165	21.8	332.9	6.2
Pump	1.7	-	3.4	-	1.7	-
Heat Exchanger	10	-	20	-	10	-
Filters, Valves, Misc.	35	-	35	-	35	-
TOTAL	1067.6	97.7	3576.4	581.8	1069.6	91.5

Total Fluid Volume = 5713.6 in^3 = 24.73 gal

TABLE 31. Baseline reservoir data

RESERVOIR	WEIGHT (lb)	INSTL VOLUME (in^3)	FLUID VOLUME (gal)
SYS. 1	17.25	442.4	1.44
SYS. 2	36.0	1579.2	5.04
SYS. 3	17.25	442.4	1.44
TOTAL	70.5 lb	2464.0 in^3	7.92 gal

2.3.5 Baseline System Weight and Volume

Weight and volume data for the baseline system is presented in Tables 32 and 33, respectively. This data is based on information contained in Appendix B. Weight data includes weight of the MIL-H-83282 hydraulic fluid.

2.3.6 Fluid Temperature

The baseline system employs heat exchangers to keep maximum fluid temperatures below +275°F. Maximum fluid temperature occurs during operation in an ambient air temperature of +110°F. Fluid temperatures used in the study for leakage, density, etc., calculations are listed in Table 34. This data corresponds to the composite study mission (average) and standard day conditions. The mission average ambient temperature is -39°F.

2.3 7 Electrical System Loads

2.3.7.1 Direct Drive Valve Electronics. Operation of direct drive valves requires the use of electronic drive units (EDU) which consume electrical power. The weight of the EDU's and their power requirements both affect fuel consumption. The incremental weight increases required in the generator and electrical distribution system were not included in the analysis. Weight and power estimates for EDU's were excerpted from previous Rockwell studies, and are shown on Table 35. Data for the conventional electro-hydraulic servo valve (EHV) design is given for comparison.

Fuel consumption resulting from use of the EDU's is shown on Figure 22. Each of the 56 dual direct drive valve (DDV) electronic packages is estimated to extract, on average, 20 watts of power. With an electrical system efficiency of 85% the power extraction will result in consumption of 24,600 lb of fuel over the life of the aircraft. The EDU's for the 56 dual valves will weigh 336 lb, and result in fuel consumption of 1.18 M-lb.

TABLE 32. Baseline system weight

	ITEM	WT/ITEM	NO.	WT/AIRCRAFT	
1.	Pump	26.0	4	104.0	
2.	Reservoir		3	70.5	
3.	Heat Exchanger		3	24.0	
4.	Filters	2.0	9	18.0	
5.	Relief valve	0.5	6	3.0	
6.	Check Valves	0.353	17	6.0	13.9%
7.	Shuttle Valves	3.5	2	7.0	
8.	Shutoff Valves	2.0	11	22.0	
9.	Pressure Reducer	1.5	1	1.5	
10.	Accumulators		4	95.8	
11.	Hand Pumps	1.8	3	5.4	
12.	Lines and Fittings				
	- F/C		1 Set	248.5	13.7%
	- Utility and Engine		1 Set	115.4	
13.	Actuation		1 Set	1926.5	72.4%
	TOTAL			2647.6 Lb	

TABLE 33. Baseline system installation volume

	ITEM	VOLUME/ITEM	NO.	VOLUME/ AIRCRAFT
1.	Pump	385	4	1543
2.	Reservoir		3	2020
3.	Heat Exchanger		3	685
4.	Filters	23	9	209
5.	Relief valve	5	6	30
6.	Check Valves	0.8	17	14
7.	Shuttle Valves	35	2	70
8.	Shutoff Valves	14.5	11	160
9.	Pressure Reducer	5	1	5
10.	Accumulators		4	1000
11.	Hand Pumps	22	3	66
12.	Lines and Fittings		1 Set	
	- F/C			2788
	- Utility and Engine			1262
13.	Actuation		1 Set	
	- F/C			7972
	- Utility and Engine			3186
	TOTAL			21010 in^3

TABLE 34. Average fluid temperature

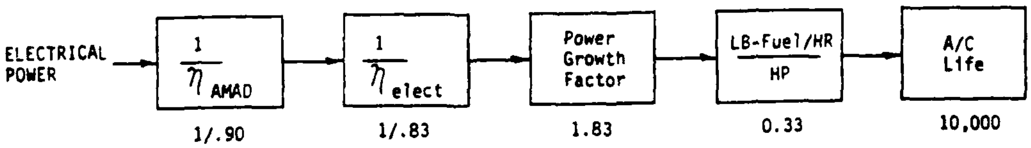
ACTUATION LOCATION	TEMP. °F
WING AND VERTICAL TAIL	80°
HORIZONTAL TAIL	180°
T/V AND ENGINE CONT.	235°
FUSELAGE	130°

NOTE:

The average ambient temperature for the mission, assuming a standard day, is -39°F. The average fluid temperature for leakage calculations was +130°F.

TABLE 35. ELECTRICAL PARAMETERS

- **Direct Drive**
  - **Two Torque Motor—Four Coils Each**
  - **25 Watt/Coil Maximum**
  - **Pulse-Width Modulated**
  - **Quiescent Amplifier Current 1 Watt**
  - **Amplifier Weight (Est) 3 Pounds**
- **Electro-Hydraulic Valve**
  - **Four Valves**
  - **0.064 Watt/Valve**
  - **Push-Pull Operation**
  - **Quiescent Amplifier Current 0.25 Watt**
  - **Amplifier Weight (Est) 0.25 Pound**



- Electrical Distribution System Efficiency 83%
- Fuel Consumption Per Aircraft Life

		Fuel, Mlb
+ Electrical Power		0.025
+ Weight		1.18
	Total	1.20

Figure 22. Fuel consumption, electrical

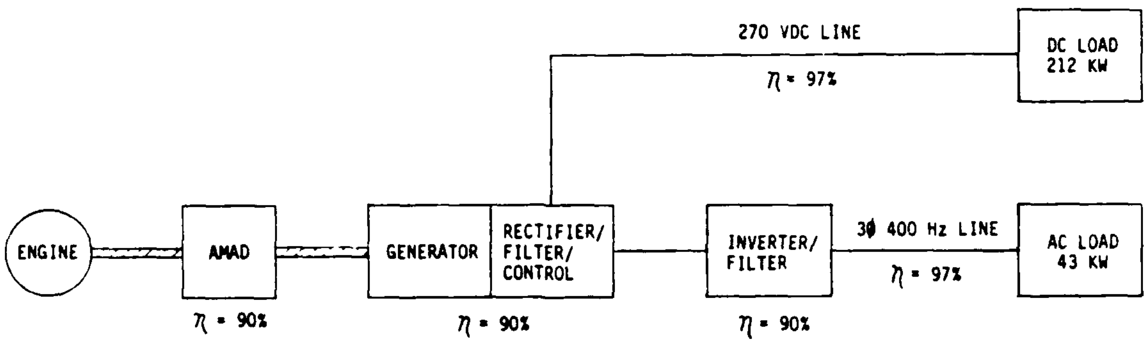
It should be noted that DDV FBW actuators are much lighter (45%) than conventional 2-stage EHV actuator designs such as used for the F-18 horizontal stabilizer. This weight saving totally overshadows the increase in electrical power caused by the EDU's. Use of direct drive valves in the baseline system was specified by the contract.

2.3.7.2 Other Loads. Estimated loads for aircraft systems other than hydraulics are presented in Table 36. Extracted horsepower from the AMAD is presented for each flight mode used in the study. This data was developed in previous Rockwell studies conducted for VFMX and ATA aircraft designs. Electrically driven fuel pumps and closed cycle ECS were assumed. Miscellaneous loads include items such as heaters, lighting, anti-ice, instruments, etc.

A typical electrical system for advanced aircraft, shown in Figure 23, was assumed for the baseline vehicle. The system has an overall operating efficiency of 77%.

TABLE 36. Estimated shaft power to electrical system in horsepower

	MISSION PHASE						
SYSTEM	TAKE OFF	CRUISE	LOITER	DASH	COMBAT	RETURN	LAND
Avionics	105	105	117	117	129	94	44
ECS	106	87	125	125	147	70	63
Fuel	84	42	42	84	84	42	78
Misc.	32	25	23	32	40	25	23
TOTAL	327	259	307	358	400	231	208



AIRCRAFT SYSTEMS TOTAL LOADING = 255 KW (342 HP)
LOSSES = 76 KW (101 HP)
TOTAL = 331 KW (443 HP)

OPERATING EFFICIENCY $= \frac{255 \text{ KW}}{331 \text{ KW}} = 77\%$

Figure 23. VSCF DC link with DC and AC outputs

2.4 ENERGY SAVING TECHNIQUES

A wide variety of energy saving techniques were investigated. They are categorized into two groups; those associated with components and those associated with systems. The techniques investigated are listed in Table 37. The following paragraphs of this section discuss the various techniques and their potential for energy savings.

2.4.1 Pumps and Integrated Actuator Packages

2.4.1.1 Pumps. The baseline hydraulic system has four 40 gpm pumps. Performance characteristics of these pumps are shown in Figure 24. Quiescent pump flow in aircraft hydraulic systems normally runs in the 10 to 20% flow range for most flight modes. Quiescent flow in the cruise mode, which dominates the baseline study mission, is particularly low. Therefore, a pump design which improves efficiency in the low flow operating range is very desirable. Efficiency at high flow is not as important since little time is spent at this operating condition. Table 38 lists the power loss areas for conventional aircraft type in-line piston pumps, and the typical percentage loss in each area at 0, 20 and 100% of rated flow. Internal leakage is the major contributor and accounts for 60 to 70% of the total losses.

Check valve pumps are widely used in industrial applications because of their ruggedness and lower cost. Use of this type of pump in aircraft has been limited, however, because of their heavier weight and relatively high pressure ripple. Advantages of the check valve pump design include a stationary non-rotating piston block, low force for variable volume control, and the ability to withstand cavitation at the inlet. A hybrid pump which capitalizes on the advantages of the in-line piston and check valve pump design features and minimizes or eliminates their respective disadvantages would be desirable.

TABLE 37. ENERGY SAVING TECHNIQUES

- **Pumps and IAPS**
- **Distribution System**
- **Accumulators**
- **Advanced Actuation**
  - Variable Displacement
  - Slimline
  - Pressure Intensifiers
- **Control Valves**
  - Aiding Load Recovery
  - Flow Augment
  - Nonlinear Valves
- **Multipressure System**
- **Hybrid Hyd/Em**
- **Advanced Materials**
- **Design Margins**
- **Thrust Vectoring**
  - Trim T/V
  - Hot Gas Diverters
- **Vehicle Control System**
  - Command Optimization
  - Variable Gain/Bandwidth

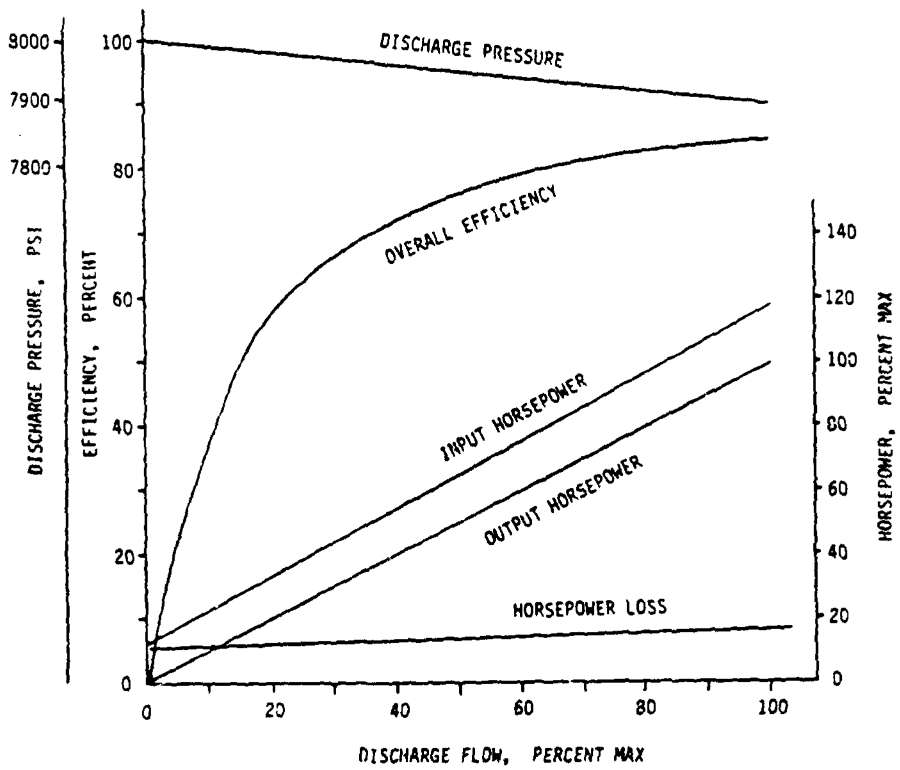


Figure 24. Baseline pump performance characteristics

TABLE 38. Conventional pump losses

Loss Area	Percentage Losses		
	0% Flow	20% Flow	100% Flow
Internal Friction	7	8	10
Internal Leakage	71	68	64
Fluid Losses	4	5	6
Fluid Compression	11	12	13
Fluid Windage	7	7	7

TABLE 39. Design comparisons

CONVENTIONAL IN-LINE PUMP	HYBRID CHECK VALVE PUMP
Rotating Barrel	Nonrotating Barrel (No Windage Losses or Centrifugal Forces)
High Compensator Forces (Pintle Bearings Highly Loaded)	Low Compensator Forces (Pintle Bearings Not Reqd)
Principal Leakage Paths: — Barrel/Port Plate Interface — Piston Shoe/Cam Interface — Compensator Control Circuit	Principal Leakage Path: — Piston Shoe/Cam Interface (Low Pressure Leakage)
Pump Cooled by Fluid Throttled From 8000 PSI	Pump Cooled by Low Pressure Inlet Fluid

A Hybrid Check Valve (HCV) pump has the potential to reduce losses in the low flow operating range. A design comparison of the HCV pump concept with the conventional inline pump is given in Table 39. Predicted performance for the HCV pump is compared to the conventional in-line pump in Figure 25. A significant reduction in losses is projected in the low flow region. Translating these savings to the 40 GPM baseline pumps results in a substantial decrease in pump losses (18 hp/pump) and a minor weight reduction. Table 40 lists the potential weight and fuel savings which would accrue by using HCV pumps in the baseline system. These savings would produce a 4.1% reduction in hydraulic system energy consumption which is equivalent to a 74 lb weight reduction.

2.4.1.2 Integrated Actuator Package (IAP). IAP's are a general class of flight control actuator which are electrically powered and have self-contained hydraulic power supplies. The IAP consists of an electric motor driven pump, reservoir, check valves, filter, relief valves, associated plumbing and hydraulic actuator. Power is supplied by the aircraft electrical system. A comparison of IAP's with conventional actuation is shown in Figure 26. The IAP can be used in place of an actuator powered by a centralized hydraulic system, or as a back-up unit in case of central hydraulic system failure. The output can be controlled mechanically, electrically (fly-by-wire), or both. Principal reasons for using IAP's are applications where 1) centralized hydraulic systems are not employed, 2) survivability is critical, 3) vulnerability to hostile ground and air fire must be reduced, and 4) small loads are located long distances from the central hydraulic power supply.

PREDICTED HCV PUMP PERFORMANCE

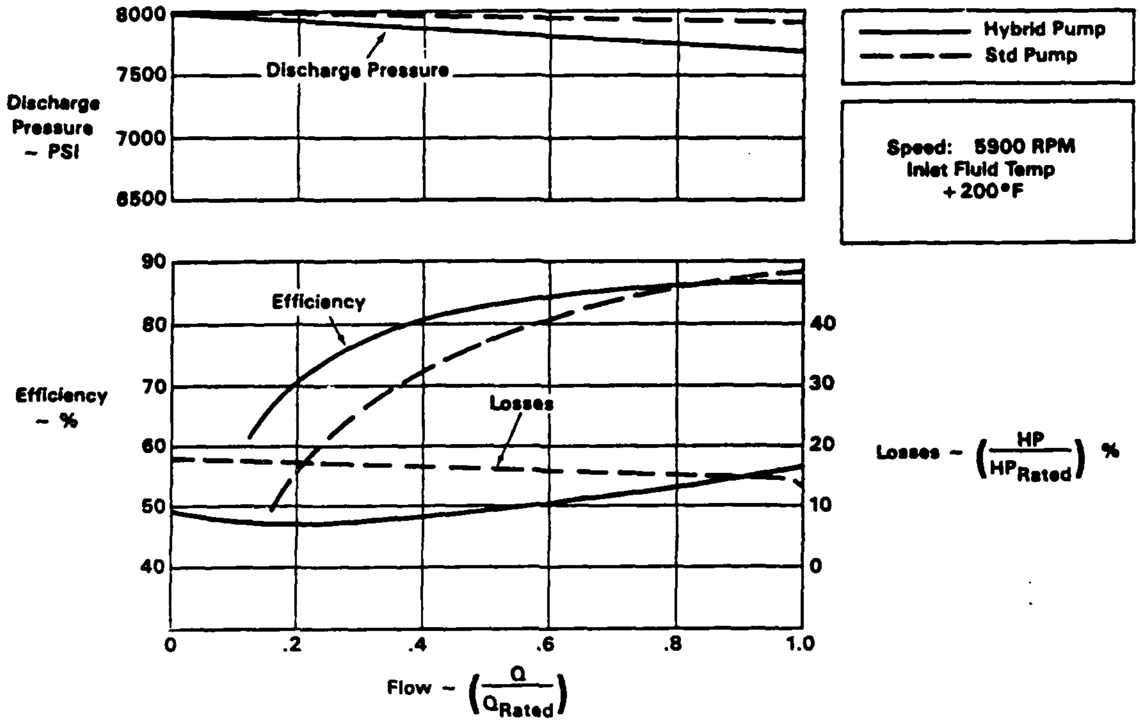


Figure 25. Predicted HCV pump performance

TABLE 40. HCV pump energy savings

• Δ Losses		– 18	Hp/Pump
• Δ Weight			
+ Pump (+6%)		+6.20 Lb	
+ Heat Exchanger		–13.00 Lb	
• Δ Fuel per A/C Life			
+ Losses		– 0.43 M-Lb	
+ Weight		– 0.02 M-Lb	
	Total	**– 0.45 M-Lb**	**(4.1%)**

There are three basic types of IAP's:

<u>Simplex</u> The simplex IAP contains a single non-redundant actuator and a single motor/pump unit.

<u>Duplex</u> The duplex IAP has two independent electric motor driven hydraulic power supplies, each providing hydraulic power for one-half of a dual tandem actuator.

<u>Triplex</u> The triplex IAP contains three independent electro-hydraulic power supplies. Two full-time power supplies provide primary hydraulic power for a dual tandem actuator, and a third unit provides emergency power in the event of a primary system failure.

IAP's have been used in many applications beginning in the 1940's. The challenge of designing IAP's for high performance military aircraft lies in dealing with severe weight and envelope penalities and in overcoming the localized heat generation/dissipation problems.

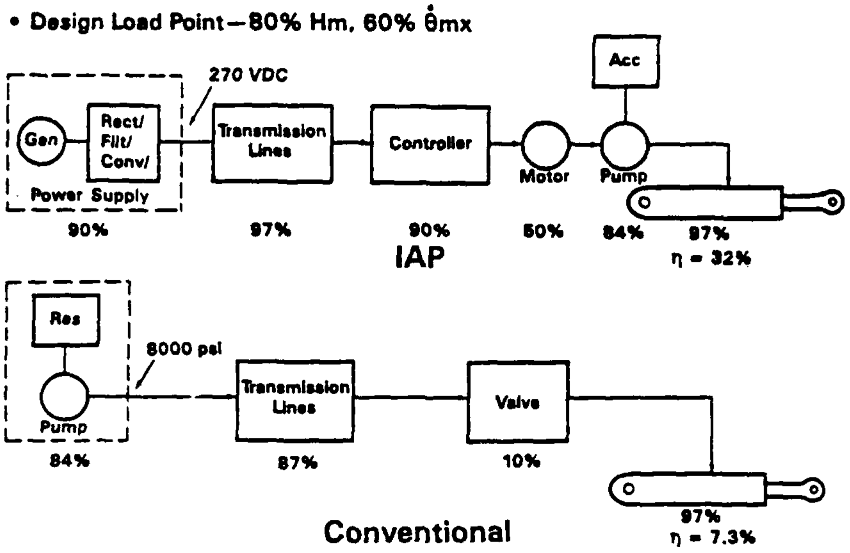


Figure 26. Efficiency comparison, IAP vs. conventional actuation

TABLE 41. IAP weights

Description	Weight, lb	Supplier	Reference
F-4 Stabilator Actuator (3000 psi, dual tandem)	38 (1)	McDonnell Aircraft	4
Simplex IAP	71 (2)	Vickers	5
	88 (3)	LTV	2
Duplex IAP	167 (4)	LTV	6
	174	GE	3
Triplex IAP	246 (est.)	LTV	6
	225 (est.)	GE	3

(1) One piece aluminum actuator body.

(2) Stripped-down F-4 actuator body. Servo pump powered one system for full performance operation.

(3) Two piece steel actuator body. Standard pump with soft cut-off powered one system for emergency operation.

(4) Does not include heat exchangers.

Several IAP development programs were conducted during the Southeast Asia conflict period of 1968 through 1971, references 2 through 6. The goal of these programs was to improve aircraft survivability from ground fire. Major participants in the programs included McDonnell Aircraft, LTV Electrosystems, Sperry-Vickers, and General Electric. Prototype simplex and duplex IAP's were built to meet the requirements of the F-4 aircraft horizontal stabilator. Laboratory and flight tests were conducted. The programs demonstrated that although performance requirements were generally met, IAP weight and volume were areas of concern. Weights of the various units are given on Table 41.

Government sponsored development of IAP's ceased after completion of the above referenced programs. A literature search covering 1972 to 1985 has disclosed no further development effort.

The application of IAP's to energy efficient hydraulic systems must consider IAP weight as a primary factor since component weight and the accompanying growth factor are directly related to aircraft fuel consumption. IAP weight, however, is partially offset by some weight savings that occur when IAP's are employed. The net change determines whether a weight advantage or penalty occurs by using IAP's.

Components Added or Re-sized	Components Eliminated or Re-sized
IAP	Servo actuator
Power wiring/control	Hydraulic transmission Lines
Electrical generator (larger)	System reservoir (smaller)
	Filters and valves (smaller)

Other factors such as IAP size (installability), redundancy, reliability, complexity, logistics, maintainability, survivability, vulnerability and cost should also be considered. However, the fundamental factor to be considered for energy savings is weight.

The net weight differentials between 1) power wiring and hydraulic transmission lines, and 2) electrical generator (larger) and hydraulic system pump (smaller) are considered minor. Reductions in reservoir, filter, and valve sizes are also considered minor. The principal weight difference occurs between the IAP and the servo actuator that it replaces. IAP's developed in the early 1970's were 2 to 4 times heavier than the original actuator. This is obviously a severe weight penalty for the benefits realized.

As a result of recent technological advances, a second look at IAP's is currently being taken. The new package is termed an electro-hydrostatic actuator (EHA). Several suppliers have built and tested state-of-the art EHA's that employ rare earth magnetic materials, electronic power switching, and microprocessor control. Design details are considered proprietary. One such unit is a 3000 psi simplex design with 9700 lb maximum output, a swept volume of 15 in^3, and a weight of 38 lb.

A direct comparison of this state-of-the-art unit with the 1970 simplex IAP's cannot be made because the 1970 IAP's had maximum outputs and strokes more than twice those of this new EHA. The weight ratio can be computed however, and compared with the earlier units, Table 42. It would appear that changes in the state-of-the-art have had little effect upon weight. The weight ratio of the 1985 unit falls within the range of the 1970 units. Results of an energy analysis applied to IAP's are summarized in Table 43. A servo pump design such as that used in the Vickers Simplex package was assumed for a duplex IAP and substituted for the baseline horizontal actuators. This unit would have a weight ratio of 3.6. As shown in Table 43, fuel consumption, due to package weight, increased significantly.

TABLE 42. IAP weight comparisons

		CONVENTIONAL (LBS)	IAP (LBS)	WEIGHT INCREASE (LBS)	RATIO (IAP/CONV)
SIMPLEX					
Vickers	1970	33	71	38	2.2
LTV	1970	33	88	55	2.7
GE	1970	33	102	69	3.0
EHA*	1985	14	38	24	2.7
DUPLEX					
LTV	1970	38	167	129	4.4
GE	1970	38	174	136	4.6
TRIPLEX					
LTV	1970	38	246 (est)	208	6.5
GE	1970	38	225 (est)	187	5.9

* Electro-Hydrostatic Actuator

TABLE 43. IAP energy analysis

	M-LBS FUEL	
	BASELINE	IAP
Weight		
Actuation (53 Lbs)	.37	1.33
Lines	.11	0.06
Leakage	.04	0.04*
Usage	.08	0.02
Generator △	.00	0.00
Pump △	.00	0.00
Total	**.60**	**1.45**

IAP for Horizontal Actuation Increases Energy Consumption 8% and Increases Weight 260 Lbs

* Assumed

There was a minor reduction due to replacement of 30.2 lbs of hydraulic lines with 16.4 lbs of wire and contactors and a slight reduction due to a higher overall efficiency (usage) of the IAP approach. The net result, however, is an increase from 0.6 to 1.45 M-lb of fuel which constitutes an 8% fuel consumption increase. System weight of the IAP configuration increased 366 lb.

IAP's and EHA's offer advantages in redundancy, survivability, and reduced vulnerability, but initial costs of ownership are high and they do not improve overall aircraft operating efficiency. IAP's are therefore not recommended for high efficiency hydraulic systems because of the weight penalty that accompanies their use.

2.4.2 Distribution System

The distribution system was designed using 5 different approaches. In design No. 1 (or baseline), the system was configured in accordance with procedures discussed in section 2.3.2 using 3Al-2.5V titanium material, and EVEN dash number tube sizes; tube data is presented in Table 44. A total distribution system weight of 364 lb, which includes fittings, clamps, and fluid was established for the baseline.

Design No. 2 utilized ODD/EVEN tube sizes: ODD for pressure lines (thick wall) and EVEN for return lines (thin wall), Table 45. This is a design method to provide a "Murphy Proof" way of using thin wall tubes for return lines. All other aspects of design No. 2 were the same as design No. 1. A total weight of 332 lb was established for design No. 2; this is a 32 lb weight savings over design No. 1.

TABLE 44. 3Al-2.5V titanium pressure and return lines, EVEN sizes

SIZED FOR BURST @ 24000. PSI
TEMP = 275.
GAMA = .0301
ULTIMATE = 112111.@ .02 WALL
MODULUS = 15.0E+06

SIZE	NOM O.D.	NOM I.D.	NOM WALL	RADIAL DEFLCT	NOM ID @ 8000	WT/FT
4	.2535	.2076	.0229	.0007	.2083	.0447
6	.3790	.3076	.0357	.0010	.3085	.1022
8	.5045	.4065	.0490	.0012	.4077	.1839
10	.6299	.5043	.0628	.0015	.5058	.2909
12	.7554	.6009	.0772	.0017	.6026	.4242
14	.8808	.6960	.0924	.0020	.6980	.5850
16	1.0063	.7923	.1070	.0022	.7945	.7603

TABLE 45. 3Al-2.5V titanium pressure and return lines, ODD/EVEN sizes

(a) Pressure Lines (ODD sizes)

SIZED FOR BURST @ 24000. PSI
TEMP = 275.
GAMA = .0301
ULTIMATE = 112111.@ .02 WALL
MODULUS = 15.0E+06

SIZE	NOM O.D.	NOM I.D.	NOM WALL	RADIAL DEFLCT	NOM ID @ 8000	WT/FT
3	.1908	.1485	.0212	.0004	.1489	.0283
5	.3163	.2577	.0293	.0008	.2585	.0705
7	.4417	.3572	.0423	.0011	.3583	.1309
9	.5672	.4556	.0558	.0014	.4569	.2342
11	.6927	.5528	.0699	.0016	.5544	.3542
13	.8181	.6487	.0847	.0019	.6505	.5011
15	.9436	.7431	.1003	.0021	.7451	.6761

(b) Return Lines (EVEN sizes)

SIZED FOR BURST @ 12000. PSI
TEMP = 275.
GAMA = .0293
ULTIMATE = 112111.@ .02 WALL
MODULUS = 15.0E+06

SIZE	NOM O.D.	NOM I.D.	NOM WALL	RADIAL DEFLCT	NOM ID @ 150	WT/FT
4	.2535	.2110	.0213	.0000	.2110	.0426
6	.3790	.3360	.0215	.0000	.3360	.0784
8	.5045	.4578	.0233	.0001	.4579	.1267
10	.6299	.5703	.0298	.0001	.5704	.1997
12	.7554	.6825	.0365	.0001	.6826	.2895
14	.8808	.7944	.0432	.0001	.7945	.3966
16	1.0063	.9060	.0501	.0001	.9061	.5211

Design No. 3 involved the use of localized velocity control to reduce water-hammer transients. A short length of larger tube size is used adjacent to fast acting valves. This reduces local fluid velocity and therefore the transient magnitude. All other aspects of design No. 3 were the same as design No. 2. This approach was found to have negligible effect on weight because 1) nearly all the tubes were sized by pressure drop requirements rather than transient requirements, and 2) the individual velocity reduction tubes (not feed lines) were fairly short in length.

Design No. 4 utilized 15-3 titanium material. A discussion of this new alloy is contained in section 2.4.8.2. Tubing wall thickness was sized for 12,000 and 24,000 psi burst pressures for return and pressure lines, respectively. Table 46 lists tube wall thickness values. The total weight for design No. 4 was established as 265 lb -- a 67 lb reduction over designs No. 2 and No. 3.

Design No. 5 utilized 15-3 titanium and reduced tube design margins. Return and pressure lines were sized for 10,000 and 20,000 psi burst pressures, respectively. The distribution system was then designed using this new size tubing, Table 47. All other aspects were kept the same as design No. 4. Only a minor weight savings of 3 lb was achieved by the use of reduced design margins. This was principally due to the requirement for a 0.020 in. minimum wall thickness to prevent denting and handling damage. Most of the return lines and a large percentage of the pressure lines were not affected by the lower design margins. Reduced margins would have a more pronounced effect on the 3Al - 2.5V titanium tubing design. Since the savings were so small, reduced safety margins are not recommended.

Fuel consumption was computed for each of the five distribution system design approaches. Since the usage fuel consumption component is small in relation to the weight fuel consumption component, and since the line loss component would be only a small fraction of the usage component, the line loss component was neglected. Fuel consumption for the distribution system designs was therefore based solely upon weight. Table 48 summarizes the 5 design approaches and lists the associated fuel consumption in terms of M-lb

TABLE 46. 15-3 titanium pressure and return lines, ODD/EVEN sizes

(a) Pressure Lines (ODD sizes)

SIZED FOR BURST @ 24000. PSI
TEMP = 275.
GAMA = .0301
ULTIMATE = 172407.@ .02 WALL
MODULUS = 15.0E+06

SIZE	NOM O.D.	NOM I.D.	NOM WALL	RADIAL DEFLCT	NOM ID @ 8000	WT/FT
3	.1908	.1485	.0212	.0004	.1489	.0293
5	.3163	.2735	.0214	.0012	.2747	.0616
7	.4417	.3887	.0265	.0018	.3905	.1134
9	.5672	.4981	.0345	.0023	.5004	.1882
11	.6927	.6076	.0425	.0028	.6104	.2819
13	.8181	.7170	.0505	.0033	.7203	.3943
15	.9436	.8265	.0585	.0038	.8303	.5257

(b) Return Lines (EVEN sizes)

SIZED FOR BURST @ 12000. PSI
TEMP = 275.
GAMA = .0293
ULTIMATE = 172407.@ .02 WALL
MODULUS = 15.0E+06

SIZE	NOM O.D.	NOM I.D.	NOM WALL	RADIAL DEFLCT	NOM ID @ 150	WT/FT
4	.2535	.2110	.0213	.0000	.2110	.0439
6	.3790	.3360	.0215	.0000	.3360	.0804
8	.5045	.4610	.0217	.0001	.4611	.1259
10	.6299	.5860	.0220	.0001	.5861	.1803
12	.7554	.7105	.0224	.0001	.7107	.2446
14	.8808	.8280	.0264	.0001	.8281	.3339
16	1.0063	.9454	.0304	.0002	.9456	.4370

TABLE 47. 15-3 titanium pressure and return lines, ODD/EVEN sizes, reduced design margin

(a) Pressure Lines (ODD sizes)

SIZED FOR BURST @ 20000. PSI
TEMP = 275.
GAMA = .0301
ULTIMATE = 172407.@ .02 WALL
MODULUS = 15.0E+06

SIZE	NOM O.D.	NOM I.D.	NOM WALL	RADIAL DEFLCT	NOM ID @ 8000	WT/FT
3	.1908	.1485	.0212	.0004	.1489	.0293
5	.3163	.2735	.0214	.0012	.2747	.0616
7	.4417	.3979	.0219	.0023	.4002	.1038
9	.5672	.5100	.0286	.0028	.5128	.1725
11	.6927	.6221	.0353	.0034	.6255	.2584
13	.8181	.7341	.0420	.0040	.7381	.3616
15	.9436	.8462	.0487	.0046	.8508	.4822

(b) Return Lines (EVEN sizes)

SIZED FOR BURST @ 10000. PSI
TEMP = 275.
GAMA = .0293
ULTIMATE = 172407.@ .02 WALL
MODULUS = 15.0E+06

SIZE	NOM O.D.	NOM I.D.	NOM WALL	RADIAL DEFLCT	NOM ID @ 150	WT/FT
4	.2535	.2110	.0213	.0000	.2110	.0439
6	.3790	.3360	.0215	.0000	.3360	.0804
8	.5045	.4610	.0217	.0001	.4611	.1259
10	.6299	.5860	.0220	.0001	.5861	.1803
12	.7554	.7110	.0222	.0001	.7111	.2438
14	.8808	.8360	.0224	.0002	.8362	.3162
16	1.0063	.9565	.0249	.0002	.9567	.4091

TABLE 48. Distribution system weight

DESIGN NO.	DESIGN			DESIGN MARGIN	WEIGHT (LBS)	FUEL (M-LBS)
	TRANSIENT CONTROL	SIZE	MATERIAL			
1	Sized by Water Hammer	Even	Ti-3Al-2.5V	24,000/24,000	364*	1.27
2	Sized by Water Hammer	Odd/Even	Ti-3Al-2.5V	12,000/24,000	332	1.16
3	Localized Velocity Control	Odd/Even	Ti-3Al-2.5V	12,000/24,000	332	1.16
4	Localized Velocity Control	Odd/Even	Ti-15-3	12,000/24,000	265	0.93
5	Localized Velocity Control	Odd/Even	Ti-15-3	10,000/20,000	262	0.92

Weight Savings Potential $\approx$ **100 Lb**

Fuel Savings $\approx$ **0.35 M-Lb**

* Baseline

of fuel per aircraft per life. The analysis shows that the distribution system weight can be reduced 100 lb by using 15-3 titanium tubing and ODD/EVEN line sizing. Considering the weight growth factor, this would lower the GTOW by 250 lb and save 0.35 M-lb of fuel during the life of the aircraft.

2.4.3 Accumulators

IR&D efforts at Rockwell have shown that helium gas has an appreciably higher energy storage capability than nitrogen (which is commonly used to charge accumulators) and is significantly higher at temperatures below 0°F. However, the low atomic weight of helium causes sealing problems. Helium gas will permeate conventional seals used in accumulators causing loss of precharge. Rockwell and Metal Bellows Corporation (recently merged with Parker Hannifin) have jointly developed an 8000 psi helium charged accumulator that successfully contains helium gas. Two units have been built and tested: an 8000 psi laboratory unit (60 cubic inches) and a larger 8000 psi unit (125 cubic inches) for an Air Force application. The advantage of helium accumulators is reduction in size and weight which translates into overall fuel savings. Less maintenance is needed since the units are hermetically sealed and do not require service during their life.

The baseline aircraft employs four accumulators; two brake accumulators, one APU start accumulator and one arresting hook accumulator. The brake and APU start accumulators are conventional stand-alone piston type accumulators and are directly replaceable with metal bellows type accumulators. The arresting hook accumulator is typically integral with the actuator. Time did not permit a design study to determine if the Metal Bellows design would be compatible with arresting hook requirements, therefore this analysis was based upon replacing only the first three accumulators. Figures B-8 and B-9 (in Appendix B) present energy storage and weight data as a function of accumulator volume. Data is given for both nitrogen and helium charged accumulators. Studies indicate nitrogen accumulators are heavier and larger than helium accumulators.

Data from Figures B-8 and B-9 were used to estimate the weight and volume of nitrogen and helium accumulators for the baseline vehicle requirements. These estimates indicate a weight reduction of 33.2 lb and a volume reduction of 138 in^3 can be achieved by using helium accumulators. This weight reduction equates to a fuel savings of 0.12 M-lb per aircraft per life. Weight, volume and energy estimates are summarized in Table 49.

2.4.4 Advanced Actuation

Four advanced hydraulic actuation concepts were investigated: rotary vane, variable displacement, slimline, and pressure intensified actuators. Rotary vane hingeline actuation was chosen for the baseline vehicle due to moldline constraints imposed by advanced aircraft, that is, very thin wings with full-time variable camber control. Trade data developed for rotary actuation in prior Rockwell studies was used as the basis for weight, volume and performance estimates presented in this section. The remaining three actuation concepts were investigated and compared against the baseline concept. The following subparagraphs discuss these concepts and their impact upon energy consumption.

TABLE 49. Metal bellows accumulators energy savings

ITEM	ENERGY STORAGE (FT/LBS)	NITROGEN		HELIUM	
		WT (LBS)	VOL (IN^3)	WT (LBS)	VOL (IN^3)
APU Start	6000	52.8	370	28.9	280
Brake					
Ground (Diff)	1689	18.0	190	12.0	155
Emergency	844	12.3	150	9.0	137
	9422	83.1	710	49.9	572

Weight Reduction—33.2 Lb

Energy Savings/Aircraft—0.12 M-Lb Fuel (1%)

2.4.4.1 Variable Displacement Actuators. Conventional fixed displacement actuators are sized for the stall load, and its control valve is designed for the maximum no-load rate. Once the actuator and valve sizes are established (by flight envelope requirements), power consumption of the actuator/valve assembly is also established. Valve losses increase as actuator load (hinge moment) decreases, reaching a maximum at no-load where nearly all power is consumed by fluid throttling across the valve, Figure 27.

One method for reducing throttling losses is to match the power capability of the actuator to the power demand of the load. This must be done on a continuous basis and requires a variable displacement actuator. One variable displacement actuator concept consists of a variable displacement hydraulic motor driving a rotary mechanical actuator. A small servo controls motor displacement (cam plate angle) as a function of load and position commands. The load sensing servo-mechanism is similar in complexity to those used in variable displacement pumps. The motor provides only sufficient torque to drive the load at the demanded rate, thus minimizing throttling losses. Figure 28 depicts a variable displacement actuator concept. This concept has been investigated in detail by Sundstrand Corporation, reference 7. The variable displacement motor approach depicted in Figure 29(a) was compared with the fixed displacement motor design shown in Figure 29(b). It was established that the variable displacement concept could save from 50 to 80% in average power consumption for specific duty cycles. Control schemes were developed and shown to provide good actuation bandwidth.

The variable displacement concept was applied to the baseline vehicle flight controls. Rotary mechanical actuators such as shown in Figure 28 were used in wing and rudder applications; a variable displacement motor driving a ballscrew actuator was used to power the horizontal tail surfaces. Leakage and weight data are compared to the baseline in Table 50. This data is based upon trade study information contained in Appendix B. Weight data for the rotary mechanical actuation concept was projected from actual values used on

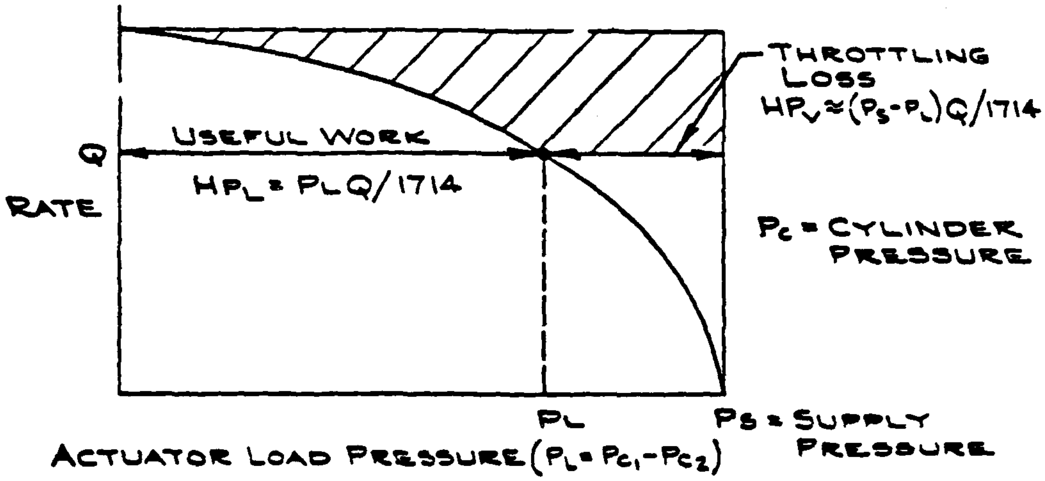


Figure 27. Control valve load/flow characteristics

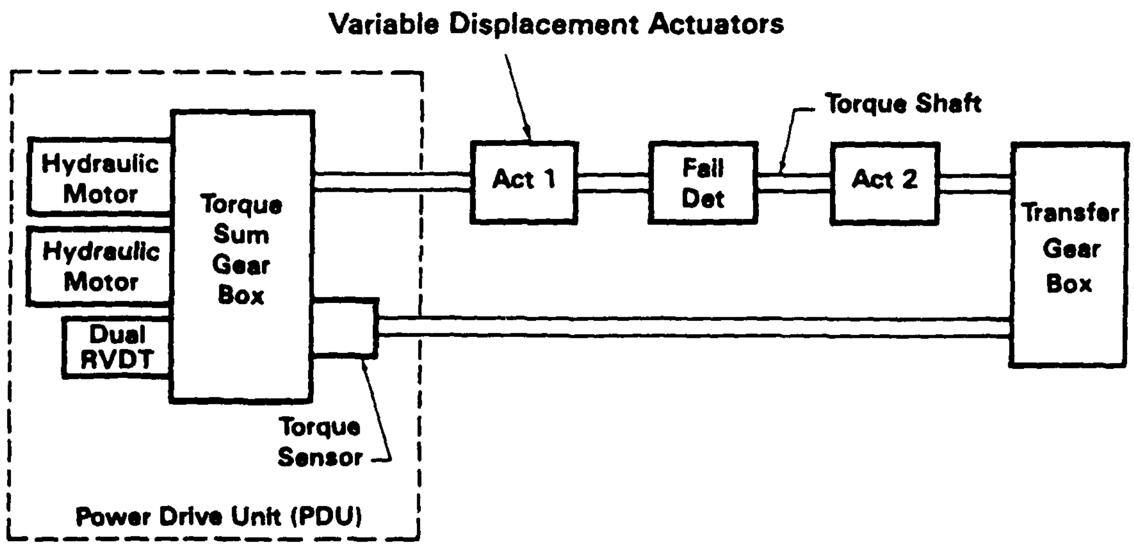
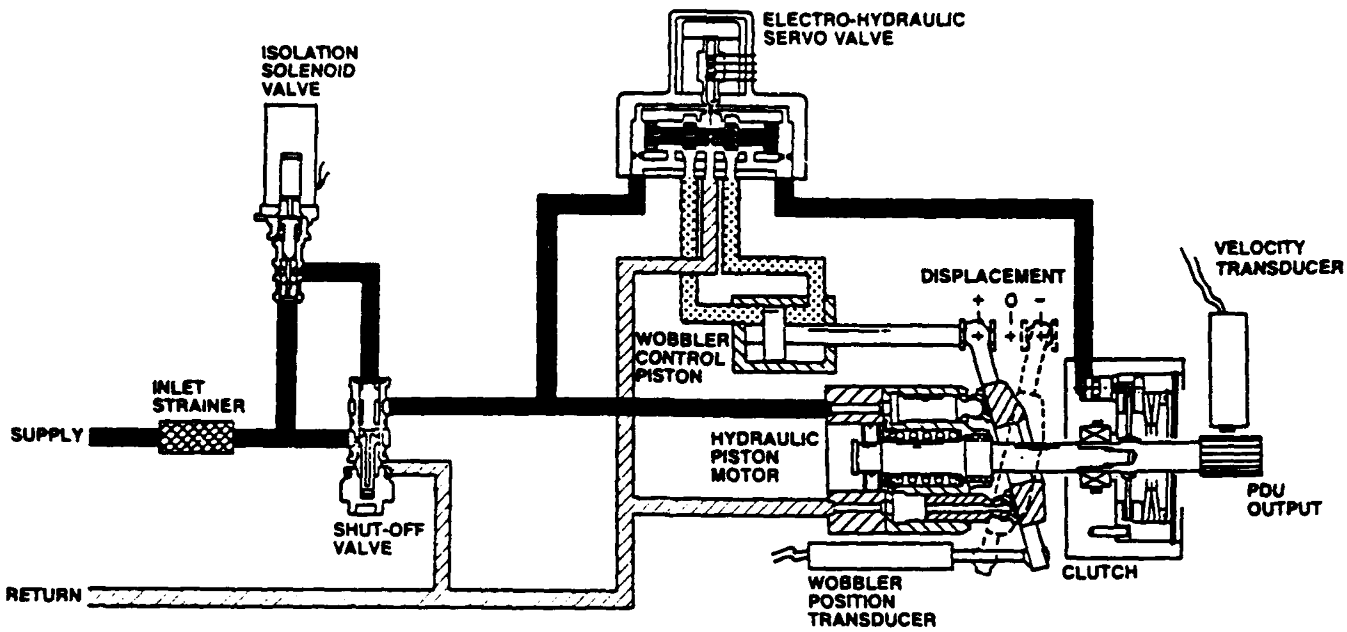
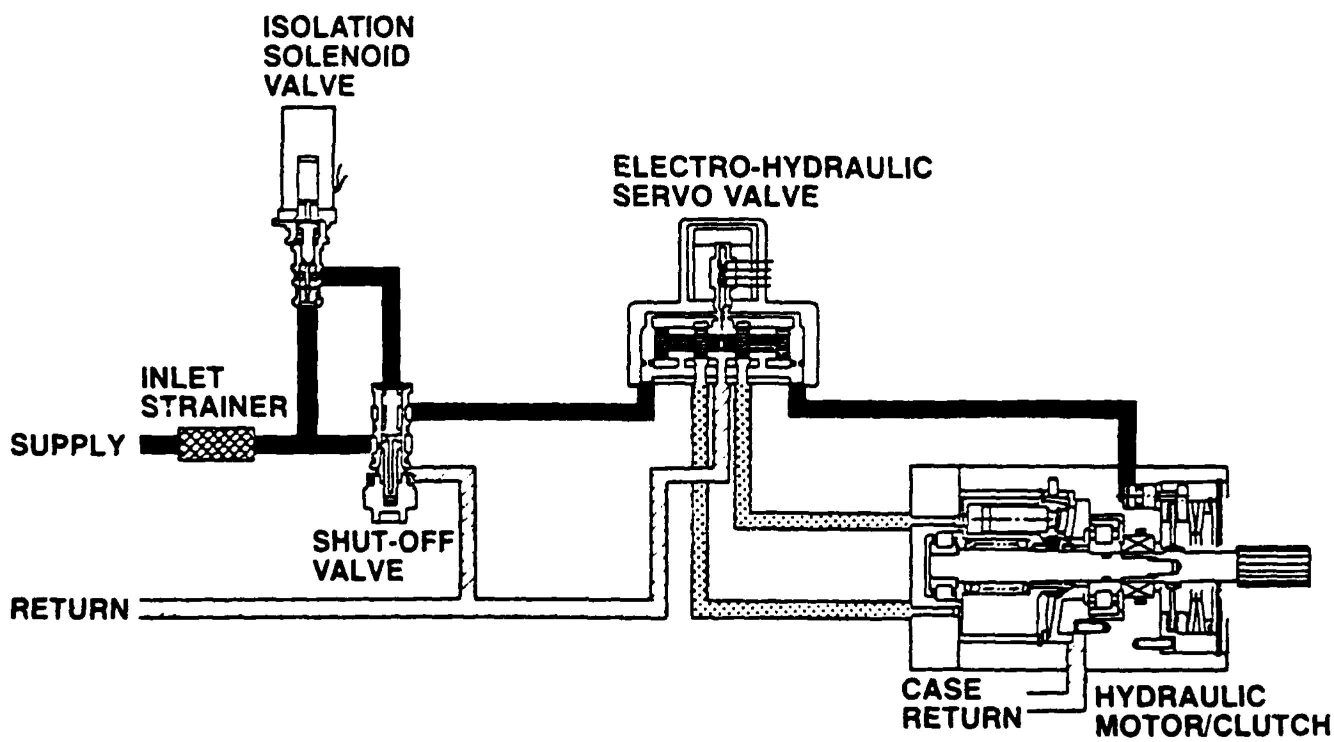


Figure 28. Variable displacement actuator concept



(a) Variable displacement



(b) Fixed displacement

Figure 29. Variable and fixed displacement motor valve schematics

TABLE 50. Variable displacement actuation, weight and leakage comparisons

SURFACE	ACTUATION TYPE	BASELINE		VARIABLE DISPLACEMENT	
		LEAKAGE/ACT (GPM)	WEIGHT (LBS)	LEAKAGE/ACT (GPM)	WEIGHT (LBS)
1 LE Outbd	Rotary	0.078	181	0.203	247
2 LE Inbd	Rotary	0.083	188	0.226	240
3 TE Outbd	Rotary	0.120	184	0.405	243
4 TE Inbd	Rotary	0.129	199	0.450	235
5 Rudder	Rotary	0.145	103	0.267	153
6 Horiz.	Linear	0.586	106	1.330	147
		3.1	961	8.3	1265

Weight = +304 Lb
Leakage = +5.2 GPM

TABLE 51. Variable displacement actuation, fuel consumption comparisons

Fuel ~ M-Lb

	BASELINE	VARIABLE DISPLACEMENT
Usage	.4784	.2545
Leakage	.2268	.3928
Pump	.7015	.7015
Weight	9.4690	10.4800
	10.880	11.8300 (+8.7%)

the B-1B aircraft rudder and wing leading edge actuation systems. Leakage for the variable displacement approach is higher than the baseline because hydraulic motors have more internal leakage than spool/sleeve type control valves.

Energy consumption was computed and is compared to the baseline in Table 51. The usage component for the variable displacement concept is reduced to about half the baseline; the leakage component is nearly double; the pump component is the same since the same size pump is required; and the weight component increased significantly. The net result is an increase in fuel consumption of 0.95 M-lb fuel per aircraft life or 8.7% higher total hydraulic system energy consumption than the baseline. The added weight of the variable displacement design completely dominates net fuel consumption. Even if the pump component could be reduced 20%, the concept would still increase fuel consumption 0.81 M-lb or 7.4%.

A second method for achieving variable displacement involves the use of a dual tandem hydraulic actuator. Displacement is changed by placing one side of the actuator in a by-pass mode. The unit would perform as a single actuator under normal operating conditions. When full hinge moment is required, both halves of the actuator would be powered. This is similar to the dual pressure level system approach except its on an actuator basis. Energy savings were computed to be 0.23 M-lb of fuel per aircraft life, Figure 30. This is 2.1% of the total hydraulic system fuel consumption. The principal disadvantage of this approach is the additional hardware and complexity required to switch from one mode to the other. The dual pressure level system concept provides comparable savings and is simpler.

- **Concept—Place One Side of Dual Actuator In Bypass Mode Except When Required by Hinge Moment**
- **Advantages—Provides Better Power Match To Load**
- **Disadvantages—Additional Complexity**
- **Potential Energy Savings**

			M-Lbs Fuel	
Weight	Δ	(0.5 Lbs/Valve)	+0.07	
Usage	Δ	(40%)	−0.21	
Leakage	Δ	(40%)	−0.09	
Pump	Δ		0.0	
		Net	−0.23	(2.1%)

Figure 30. Energy savings, dual actuator by-pass mode

2.4.4.2 Slimline Actuators. The trend in advanced aircraft is toward thin wings witn more camber control to improve the lift-to-drag ratio ($L/_D$) over the entire flight envelope. Thin wings were assumed in the baseline vehicle to be representative of advanced technology, Figure 31. Advanced actuation studies conducted by Rockwell in support of the ATF and NASP programs have shown hydraulic rotary vane actuators are the best solution for thin wing flight control installations. Consequently, rotary vane actuation was assumed for leading and trailing edge flap and rudder actuation in the baseline design.

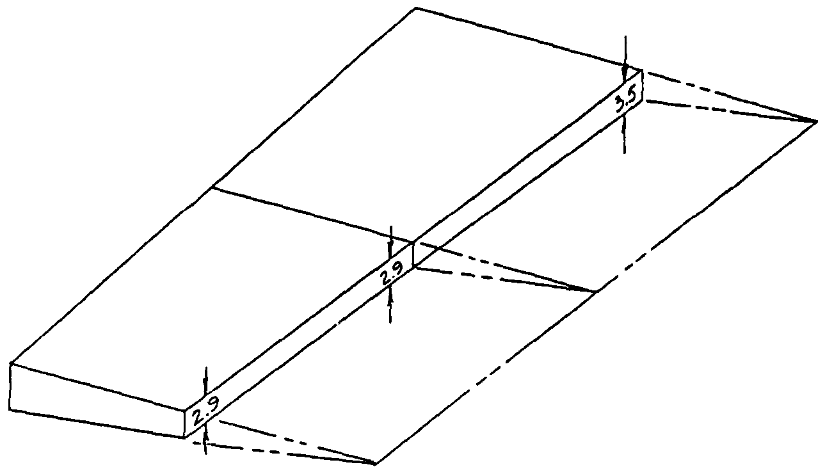


Figure 31. Baseline wing torque box thickness

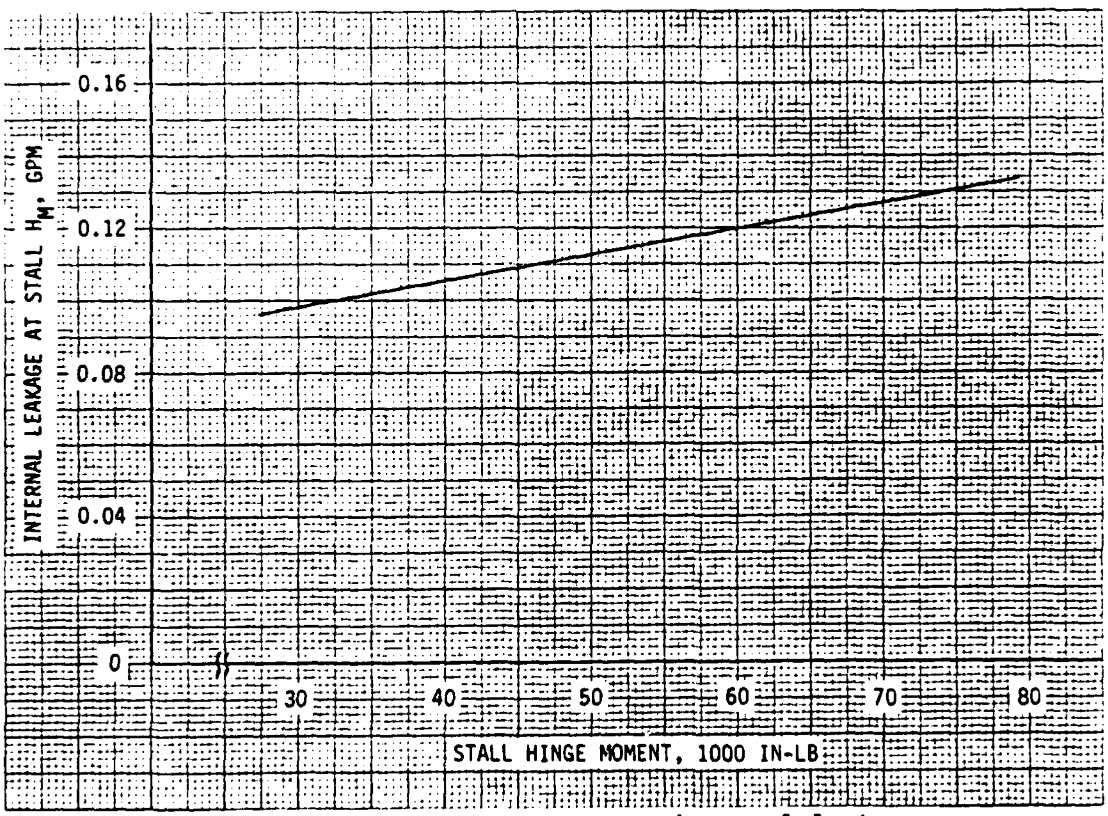


Figure 32. Vane actuator internal leakage

"Slimline" is a term coined by Rockwell to describe a class of advanced, low-profile, 8000 psi flight control linear actuator which has the control valve mounted in an axial location rather than on the side. Two configurations were evaluated for the trailing edge flap application in the baseline aircraft: 1) a conventional bellcrank arrangement which requires wing pods to house the actuators; and 2) a hingeline installation which uses a mechanism to convert linear piston motion to angular control surface motion. These installations are depicted in Figures 33 and 34. Preliminary designs were conceived for both installations to establish the kinematics, size and weight of each.

The pod installation has a bellcrank arm length of 4.4 inches. This was found to be the best compromise between size, weight and hinge free-play (resolution). Pod envelope dimensions to accommodate this design are given in Table 52. Eight pods are required. Energy consumption for the pod installation was computed and is compared with the baseline in Table 53. Weight of the pod design was estimated to be 76 lb less than the baseline. A weight estimate breakdown is given in Table 52. Linear actuator weight alone is considerably less than the rotary vane actuator, however, when actuator supports, bellcranks and pods are included, the difference is reduced. Weight reduction saves energy. Linear actuators do not have as much internal leakage as rotary vane actuators; this also saves energy.

A limited amount of data is available on vane actuator seal leakage. Several suppliers are developing rotary actuation for thrust vectoring on engines and for vane actuation. Based upon limited data from such programs, actuator leakage at stall versus stall hinge moment (actuator size) was estimated and is presented in Figure 32. This data was used to compute the fuel consumption due to leakage for dynamic loading and found to be insignificant because the average differential pressure is quite low. Steady state loads result from trim conditions. Longitudinal trim is accomplished by the horizontal stabilizer which utilizes a linear actuator.

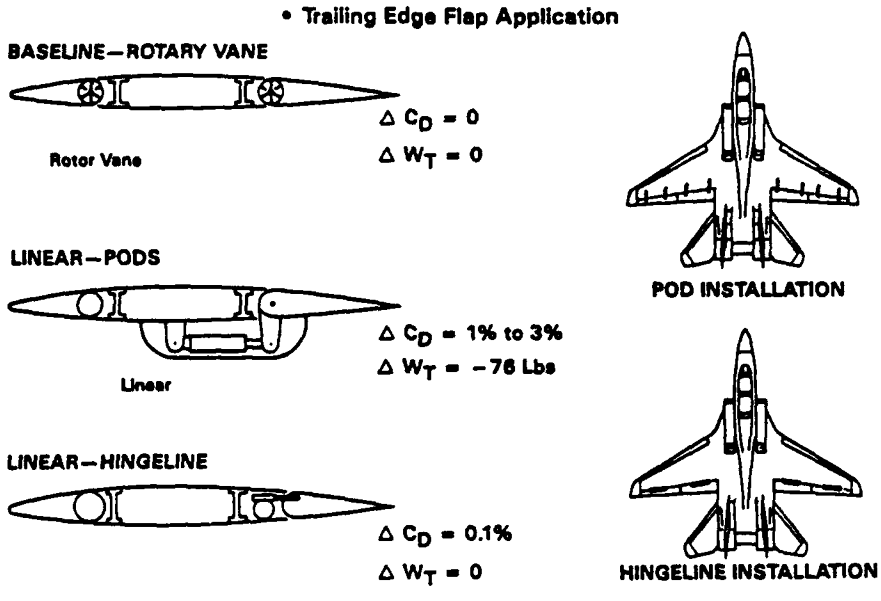


Figure 33. Slimline actuation

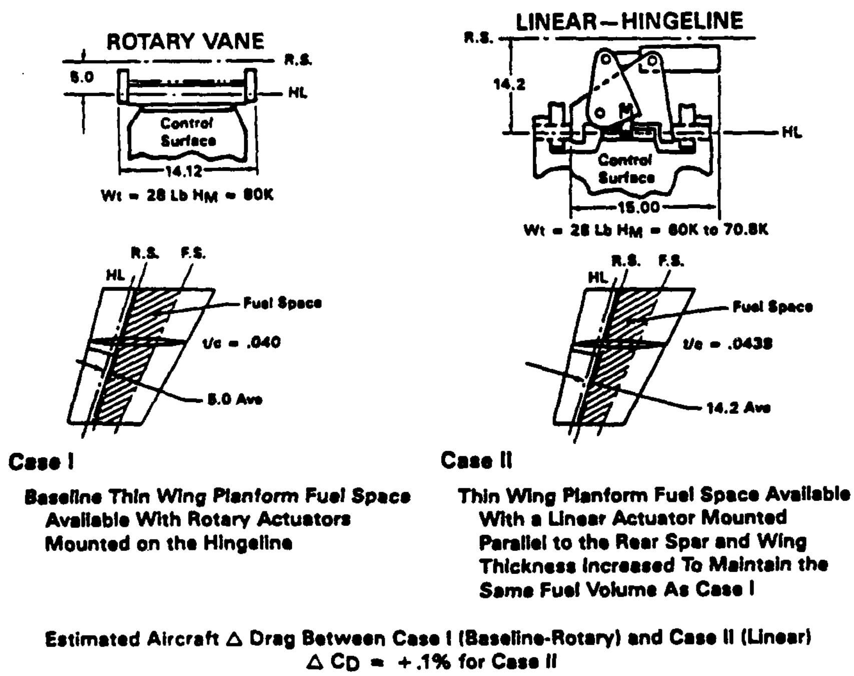


Figure 34. Thin wing fuel space availability

TABLE 52. Pod design weight estimate

POD		ASSEMBLY	
Bell crank	2.05 lb	Pod	9.5 lb
Supports	2.05	Actuator	22.0
Bolts	1.43	Fluid	1.6
Fairing	3.97	Support	6.4
Total	9.5 lb	Total	39.5 lb

Baseline Actuator Weight 49 lb

Total Weight Difference

= 8 x (49 -39.5) = 76 lb

Pod Size 4.5" thick by 6.8" wide by 50" long

TABLE 53. Linear vs. rotary energy comparisons

	Δ FUEL TO BASELINE	
	LINEAR–(POD) (M-LBS)	LINEAR (HINGELINE) (M-LBS)
Δ Usage	—	—
Δ Leakage	–.0004	–.0004
Δ Pump	—	—
Δ Weight	–.2660	—
Δ Drag	+.7180	+.0770
Net	+.4516 (+4.2%)	+.0766 (+.7%)

Lateral and directional trim conditions are minimal except for asymmetric stores conditions which, on a percentage basis over the life of the aircraft, occur infrequently and do not produce significant energy losses. The leading edge flap actuators carry a steady state load which varies as a function of Mach number. An estimated mission average load for the leading edge flaps is 10% of stall. Based upon this load and the leakage characteristics in Figure 32, 3130 pounds of fuel will be consumed by the leading edge actuators; 150,000 pounds of fuel consumption results from servo valve leakage.

2.4.4.3 Pressure Intensified Actuators. The pressure intensifier actuation concept is compared with a conventional design in Figure 35. In essence, a pressure intensifier is placed upstream of the control valve. During low hinge moment load conditions the intensifier is "off"; at high loads the intensifier is "on" and pressure to the control valve is boosted 150%. In an 8000 psi system, the actuator/valve would be designed to operate at 12,000 psi. The higher pressure is employed only to meet the stall hinge moment requirement. Smaller actuators require less flow to produce the same surface rate. Thus, during low load conditions (which occur during the majority of flight time), less power would be extracted from the engines than with conventional actuation. The system should be more efficient because the actuator/valve design more closely matches the load over the operating envelope and less power is wasted in the control valve.

An actuation system must be designed to meet all requirements, thus in the conventional system, actuator displacement would be based on stall hinge moment and supply pressure, and the valve would be sized by the maximum no-load rate requirement ($\dot{\theta}_{MAX}$). The operating envelope of this approach is labeled "Conventional Design," Figure 35. The design-load point is met since it is within the envelope. The three design points can also be met with the Pressure Intensifier (PI) approach. Sizing the actuator to provide 2/3 stall hinge moment at supply pressure results in a smaller actuator which requires only 2/3 of the flow to meet the no-load rate requirement. This approach

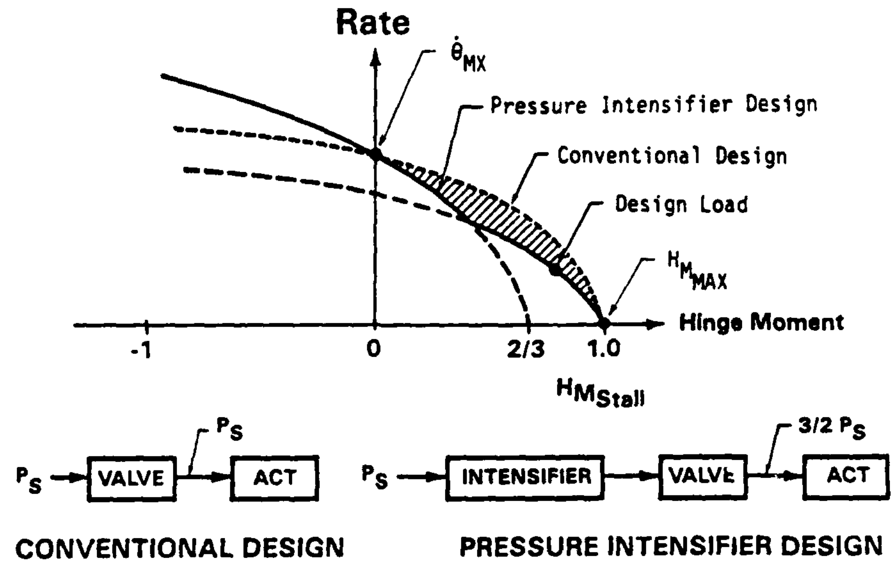


Figure 35. Pressure intensifier approach

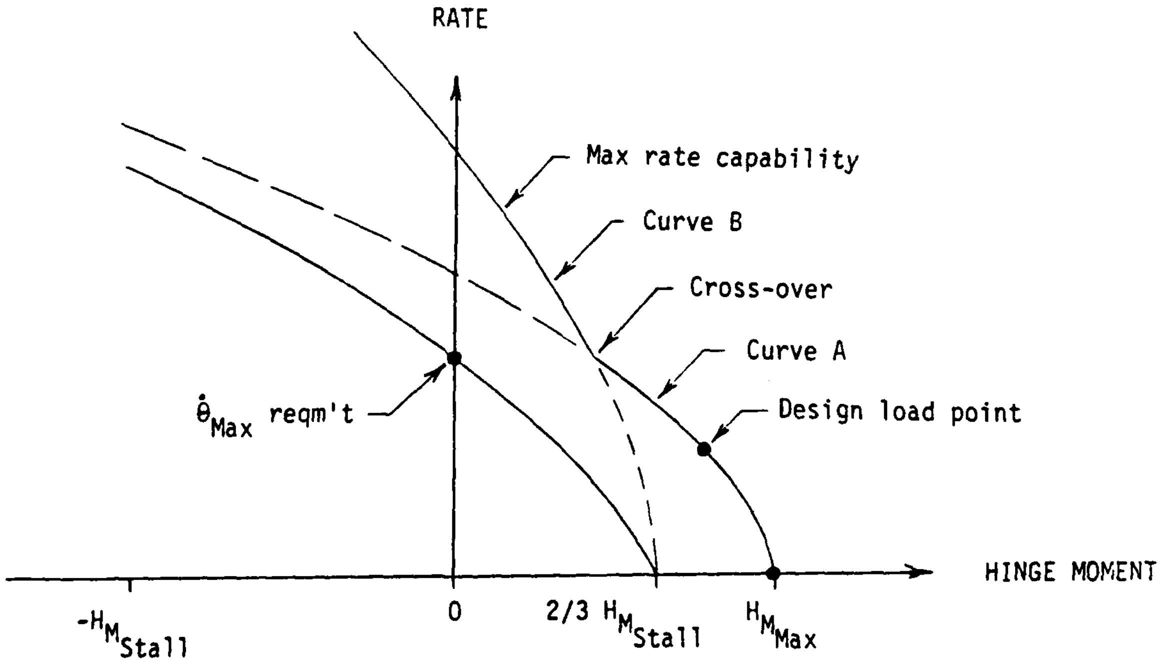


Figure 36. Pressure intensifier design curves

also provides greater rate capability in the negative load region (aiding loads). The stall hinge moment requirement $(H_{M\,MAX})$ is met by the boost pressure supplied by the pressure intensifier. The control valve is sized by the nc load rate requirement.

Baseline system requirements are such that the valve is sized by the design-load point rather than the no-load rate requirement, as depicted by curve 'A' in Figure 36. The pressure intensifier concept can still be employed to save "usage" energy. The valve/actuator/PI would be sized to provide the performance indicated by curve B. At pressures below the crossover point the rate capability is higher than necessary and could be limited by a restrictor or other means.

The energy per cycle extracted from the hydraulic system for a periodic command has been shown to be equal to

$$W_I = 4 P_S D_m A$$

and is linearly dependent upon actuator displacement. The PI design reduces actuator displacement and, therefore, reduces the extracted energy/cycle. For the 2/3 reduction in D_m design considered previously, the energy per cycle is decreased by 1/3 when operating below the crossover point. The total energy savings must include the efficiency of the PI and its incremental weight change.

The total energy savings afforded by the PI approach was estimated based upon the following:

1. 2/3 reduction in D_m
2. Actuator weight vs pressure data in Appendix B
3. Pressure intensifier weight data in Appendix B
4. Replace F/C, convergent flap, and T/V flap actuators with the PI design equivalent. Engine and utility actuators were the same as the baseline design.

Weight and energy estimates are presented in Figure 37. Actuator weight increases nearly 15% or 178 lb. The design utilizes 24 pressure intensifiers which increases weight by an estimated 215 lb. Hydraulic distribution lines and pumps were assumed to be the same as the baseline since the design load point flow requirements are the same. A net increase in fuel consumption of 1.17 M-lb or 11% was computed for the PI design; weight increased 393 lb. If the baseline design load point requirements were lowered such that maximum power demand was set by $\dot{\theta}_{MAX}$, a reduction in pump size and lines would be possible which would reduce consumption. This, however, could never overcome the increase due to actuator and PI weight.

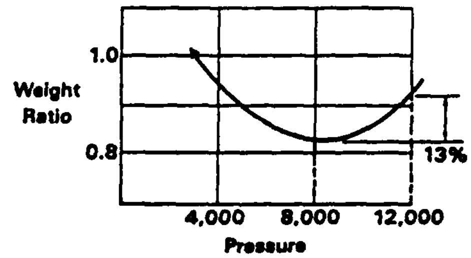
Results of the study are summarized in Figure 38. The concept has no energy saving potential, requires 12,000 psi technology development, and increases complexity. Finally, the dual pressure level system is a better approach to load matching.

PRESSURE INTENSIFIER—ENERGY ANALYSIS

Application: Primary Controls

Weight

Actuation	Δ	178	Lb
Intensifier	Δ	215	Lb
Line	Δ	0	
Pump	Δ	0	
	Total	393	Lb



Energy	**M-lb**
Usage Δ	−0.15
Valve Leakage Δ	−0.06
Pump Δ	0.00
Weight Δ	+1.38
Net	+1.17

Pressure Intensifiers Increase Energy Consumption 11% and Increases Weight 393 Lbs

Figure 37. Pressure intensifier energy analysis

PRESSURE INTENSIFIER SUMMARY

ENERGY SAVINGS POTENTIAL MINIMAL

- (Lb-Fuel/Lb-Wt) Dominates

REQUIRES 12,000 PSI TECHNOLOGY DEVELOPMENT

- Seals
- Intensifiers
- Actuators and Valves

ADDED COMPLEXITY

- 24 Intensifiers
- Actuation Control

DUAL PRESSURE LEVEL SYSTEM IS BETTER APPROACH TO LOAD MATCHING

Figure 38. Pressure intensifier summary

2.4.5 Control Valves

A major portion of the energy consumed in operating primary flight control actuators is in control; very little energy is expended in performing actual work. The potential for saving energy is, therefore, high in the control element, i.e., the valve. Power loss in spool/sleeve type valves can be divided into two categories: 1) quiescent or leakage losses and 2) operating or throttling losses.

Quiescent losses for a typical control valve are depicted in Figure 39. Losses are highest at null and decrease as the spool moves away from null. Although leakage is relatively small in direct drive valves, it is continuous regardless of whether or not the actuator is used. In fact, internal leakage is highest when the actuator is not being used. Quiescent leakage, therefore, causes appreciable power loss. Operating losses increase further due to the need for servo control of actuator position.

The spool/sleeve valve "throttles" fluid when controlling actuator rate. This is depicted in Figure 40. For example, to move the load (P_L) at a given rate (Q_L) the valve must provide a pressure drop of (P_S-P_L) to reduce the supply pressure (P_S) to the required load pressure. This pressure drop (throttling) is a power loss, i.e.,

$$H_P = \frac{(P_s - P_L)\, Q_L}{1714}$$

Techniques to reduce the power loss in both categories are listed below and discussed in the following subparagraphs of this section.

Technique	Category
Non-Linear Valves	Quiescent
Aiding Load Recovery Valves	Operating
Flow Augmentation Valves	Operating

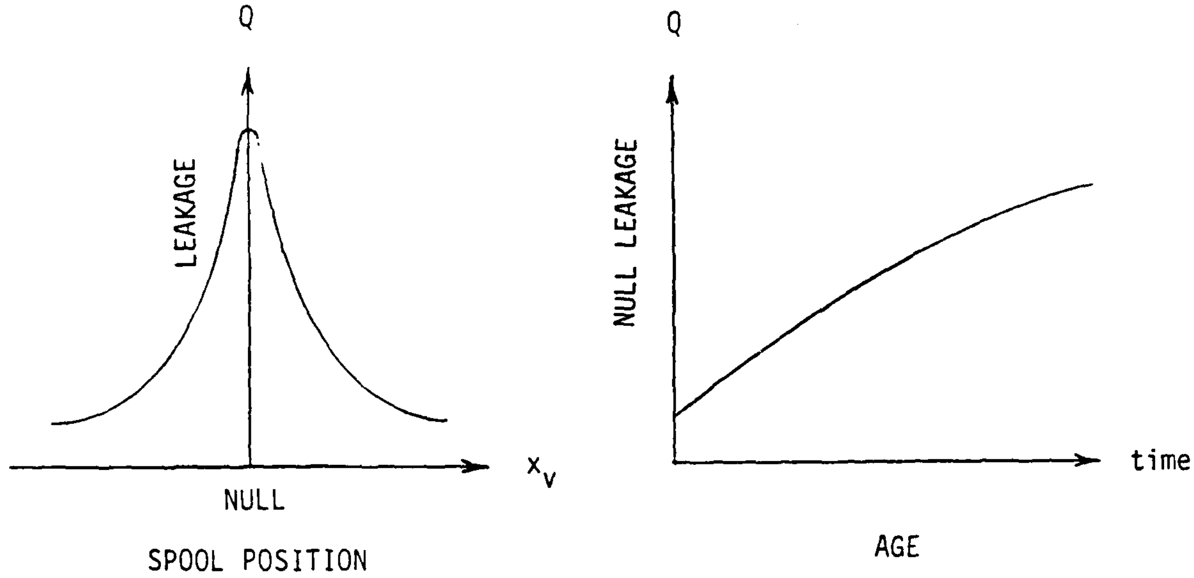


Figure 39. Typical control valve leakage

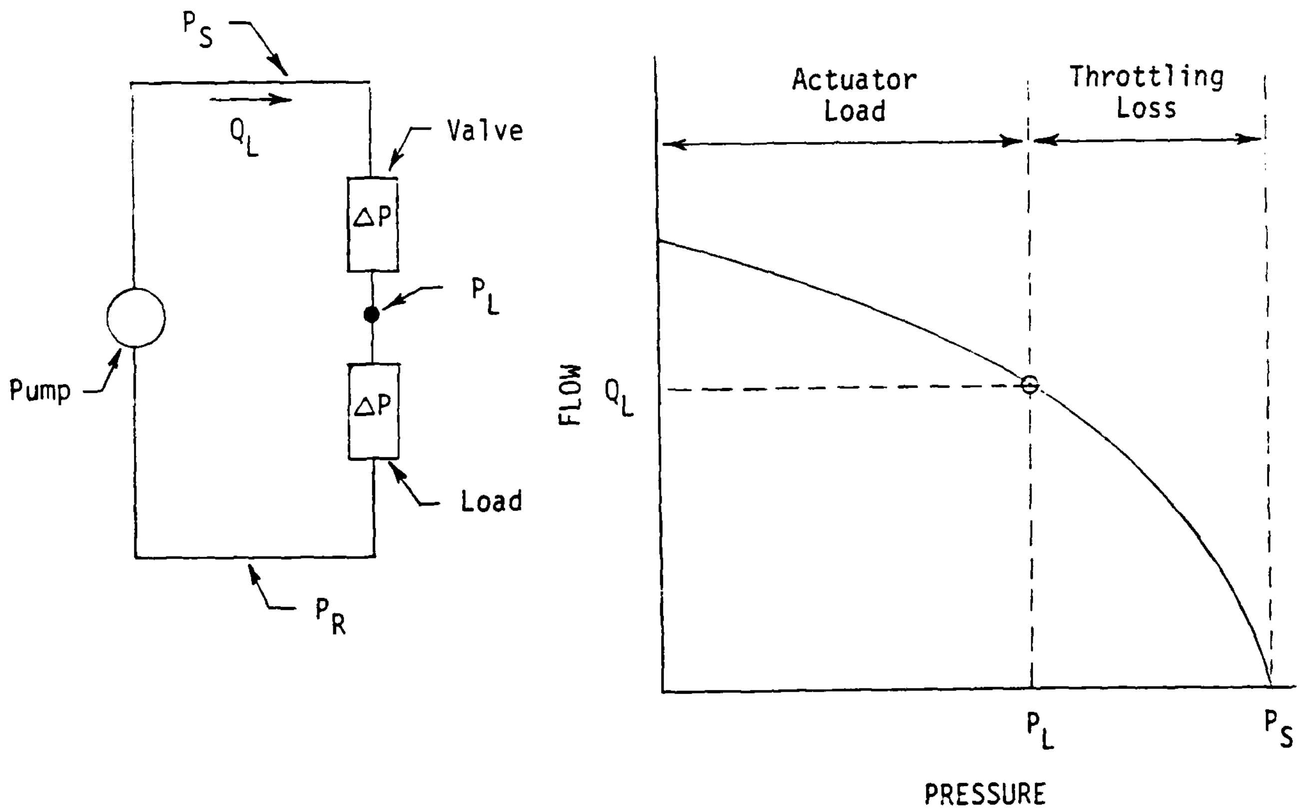


Figure 40. Control valve throttling

2.4.5.1 Internal Leakage. The baseline system has 56 dual direct drive servo valves with spool/sleeve type control elements. The valves were sized based on criteria given in Table 23. Internal leakage is a function of spool position as shown in Figure 39. Null leakage is a function of valve size (no-load flow), internal clearances, orifice design (shape, overlap, etc), and wear. The average leakage flow for a valve over the aircraft life was determined by:

$$C_{Ave} = C_{Null} \times C_{Wear} \times C_{Dynamic}$$

Where, C_{Null} = Null leakage of valve in new condition

C_{Wear} = Factor to account for wear during valve life

$C_{Dynamic}$ = Factor to account for the fact the spool moves dynamically and is not always at the null position.

Null leakage in new valves was estimated from the "average quality" trend line shown in Figure 41 and is based on empirical data acquired in the LHS and VHP test programs. Specific data points are plotted. Wear data was obtained from endurance tests conducted in the same programs and is presented in Figure 42. Available data was limited to 1200 hours of usage. It was necessary to extrapolate the curve to the 10,000 hour baseline vehicle design life. Each point in Figure 42 represents the average of 12 valves. The accuracy of this extrapolation of leakage as a function of time is uncertain. Energy loss due to leakage was based upon this trend and, therefore, has this uncertainty.

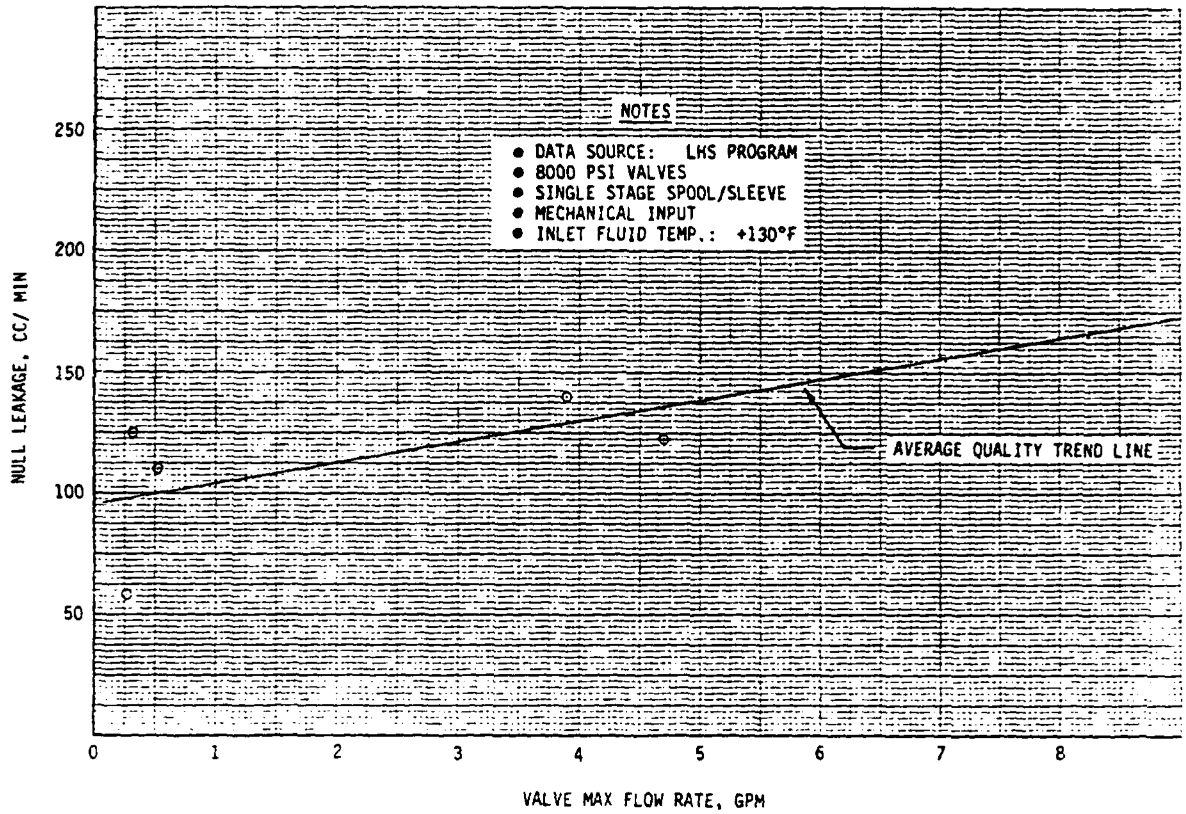


Figure 41. <u>Control valve leakage vs. size</u>

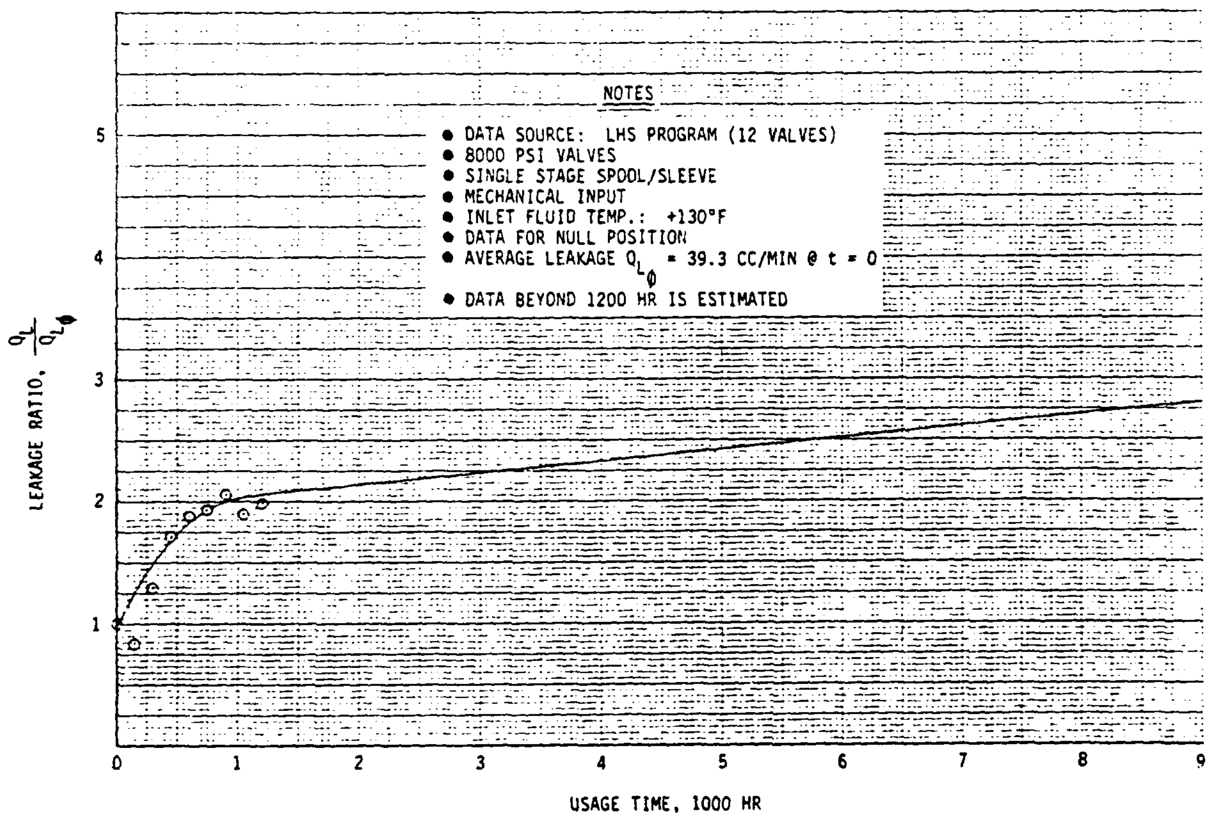


Figure 42. <u>Control valve leakage vs. time</u>

An average wear factor (C_{Wear}) of 2.16 was assumed for this data. A dynamic factor ($C_{Dynamic}$) of 0.54 was established by computing the leakage per cycle, summing over the total number of cycles per A/C life, and combining this loss with the leakage that occurs during idle time. Details of these calculations are presented in section 2.4.5.1.1.

Fuel consumption per aircraft life due to valve leakage was computed for each valve based upon the average leakage (Q_{Ave}) value. This data is contained in Table 54. Power extracted from the hydraulic system was computed by:

$$HP = \frac{Q_{AVE}\, P_S}{1714}$$

Shaft extraction power was then calculated by dividing by the AMAD efficiency (η = .9). (See Figure 22)

Pump efficiency is highly dependent upon its operating point, and varies from 0 at no flow to 85% at full output flow. Data from the LHS program indicates that quiescent pump output typically ranges from 20 to 30% with transient demands to 100%. A quiescent flow of 25% was selected; average pump efficiency would, therefore, be 64%. Pump losses are relatively independent of the power extraction level, Figure 24. Losses range from 12 to 18 percent of rated output as output varies from 0 to 100%. At the selected quiescent point, pump losses are 14 percent of rated output. Pump inefficiencies could be handled in the calculations by an efficiency number or as a separate loss component. Since pump losses are essentially constant, the separate loss component method was used. Fuel consumption resulting from valve leakage, Table 54, therefore, excludes pump efficiency. This is also true for all data presented elsewhere in this report.

TABLE 54. Fuel consumption due to valve leakage

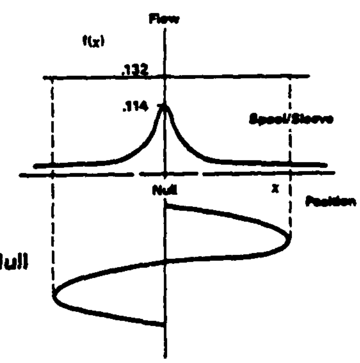
o AIRCRAFT LIFE 10,000 HOURS

LOAD*	LB FUEL/AIRCRAFT-LIFE
1	28110
2	31240
3	32410
4	36010
5	24010
6	79220
7	70050
8	140100
9	18900
10	1668
11	625
12	2085
13	1582
14	6081
15	4343
16	1689
	226800 LB

* See Tables 17 and 18

$$Q_{Avg/Cyc} = \frac{1}{\pi}\int_{0}^{\pi} f(A \sin \omega t)\, d\,\omega t$$

$$Q_{Avg/Life} = \left[Q_{AveCyc}\left(\frac{T_{Cyc}}{T_{Life}}\right) + Q_{Null}\left(\frac{T_{Life} - T_{Cyc}}{T_{Life}}\right)\right] \approx .54\, Q_{Null}$$



EHV

1st Stage	.528 GPM
2nd Stage	.070 GPM
	.598 GPM

DDV

Spool/Sleeve	.211 GPM

Figure 43. Control valve dynamic leakage

2.4.5.1.1 Dynamic Leakage. - Valve leakage is maximum at null and decreases as the spool is displaced from null, Figure 43. The average leakage during an actuation cycle was estimated by dividing the volume of fluid leaked by the time period.

$$q = f(x)$$

$$x = A \sin \omega t$$

$$Q_{AVE}/cyc = \frac{2}{T}\int_0^{T/2} f(A \sin \omega t)\, dt$$

$$= \frac{1}{\pi}\int_0^{\pi} f(A \sin \omega t)\, d\omega t$$

The average leakage per cycle is thus a function of spool displacement, amplitude, and frequency. The actuator usage function defines the amplitude and number of cycles for each actuator. Frequencies used are given in Table 12. Spool displacement was determined by solving the following set of equations

$$P_L D_m = J\ddot{y} + B\dot{y} + Ky$$

$$q = Q_{NL}\frac{x_v}{x_s}\sqrt{1 - \frac{P_L}{P_S}}$$

$$q = D_m \dot{y}$$

$$y = A \sin \omega t$$

$$\dot{y} = A\omega \cos \omega t$$

$$\ddot{y} = -A\omega^2 \sin \omega t$$

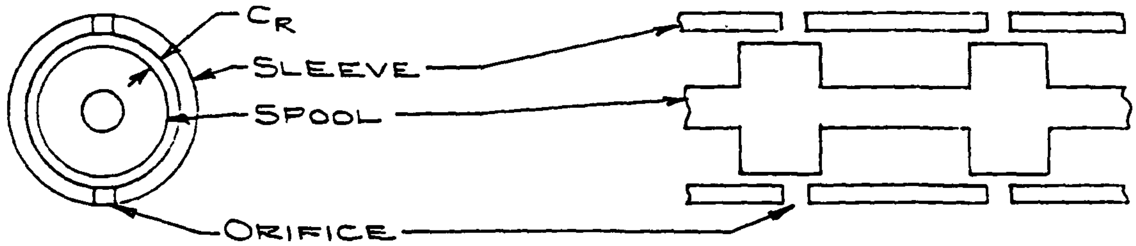
and integrating over the time period to obtain average leakage.

The total leakage volume is the sum of the dynamic leakage while the actuator is cycling plus the null leakage while the actuator is stationary. The frequency assumed in the calculation was the aircraft natural frequency for the particular control axis. Cycle time (T_{cyc}) is then the product of the total number of cycles and the cycle period. This computation is illustrated in Figure 43.

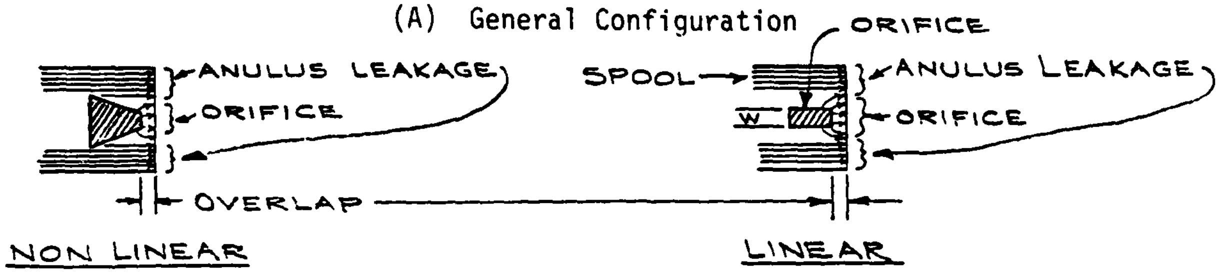
2.4.5.1.2 Non-Linear Valves. Non-linear control valves were studied to determine their potential for reducing quiescent leakage losses and thereby save energy. Two types of valves - - high overlap and "V" shaped orifice - - were investigated. Their potential for saving fuel was assessed and found to be significant; however, the non-linearity introduced in the control loop is undesirable and requires special compensation. (Valves are termed linear or non-linear based on their no-load flow characteristics.) Details of this study are presented in the following paragraphs.

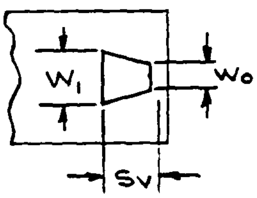
Radial clearance between the valve spool and sleeve is typically held to 100 to 125 micro inches to minimize leakage and to prevent silting. Orifice width is based on flow requirements and spool stroke. Valve overlap is determined by 1) the importance of minimizing null leakage, and 2) the accuracy with which the orifices can be located during manufacture.

Leakage flow in spool/sleeve valves consists of two components: 1) flow in the annulus (clearance) between the spool and sleeve, and 2) leakage in the orifice area. Valve geometry and leakage components are illustrated in Figure 44. Typically, the orifice component is one to two orders of magnitude larger than the annulus component and determines valve null leakage. Equations for computing the leakage components are presented in Figure 44. Valve parameters which determine leakage are radial clearance, orifice width and shape, and valve overlap. Too little clearance can cause spool "sticking" due to "silting". During periods of low activity, fluid particulate contaminants (silt) wash into and remain in the annulus between

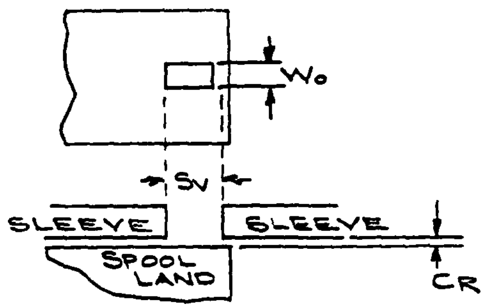


(A) General Configuration





(B) 'V' Shape Non-linear Orifice



(C) High Overlap Non-linear Orifice

ANNULUS LEAKAGE

$$Q = \frac{\pi R C_R^3}{6 \mu L} \left[1 + \frac{3}{2} \left(\frac{e}{C_R} \right)^2 \right] \Delta P$$

ORIFICE LEAKAGE

$$Q = \frac{\pi W C_R^2}{32 \mu} \Delta P$$

RECTANGULAR PASSAGE FLOW

$$Q = \frac{W C_R^3}{12 \mu L} \Delta P$$

WHERE,

Q = FLOW - IN^3/SEC
R = SPOOL RADIUS - IN
C_R = RADIAL CLEARANCE - IN
μ = FLUID VISCOSITY
L = ANNULUS LENGTH - IN
W = ORIFICE WIDTH - IN
ΔP = PRESS. DIFF - PSI
e = ECCENTRICITY - IN

Figure 44. Control valve leakage parameters

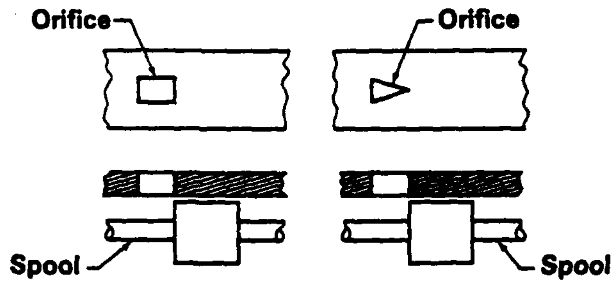
the spool and sleeve. This "silt" builds up with time and can cause the spool to stick in the null position after a period of idleness. Spool/sleeve clearance in the 100 to 125 micro inch range is a good compromise between minimizing leakage and avoiding silting.

Orifice leakage is a linear function of orifice width. Reducing orifice width reduces leakage, however, the width parameter in rectangular orifice design is limited to a minimum value which provides the required valve no-load flow at full spool displacement. Valve overlap is very effective in reducing leakage but introduces deadband in the control loop which is undesirable.

V-Shaped Orifice

The orifice leakage equation (Figure 44) gives the laminar flow through a sharp edge rectangular slit (zero overlap) which is the flow passage area between the orifice and the spool land edge. Leakage is directly proportional to the orifice width (W). Reducing W reduces leakage and when carried to the limit, results in a V-shaped orifice. The orifice leakage component becomes quite small but not zero as implied by the equation. In the vicinity of the V apex, flow is better described by the equation for flow through a short tube orifice. Further from the apex, the rectangular passage-way equation better describes the flow. The average orifice width is an inverse function of valve stroke (S_V). Valve stroke should , therefore, be as long as possible to minimize leakage. DDV strokes typically range from 0.005 to .030 in.

Annulus and orifice leakage components were computed for two 5 gpm critical center valves which were identical except for the orifice shape; one was rectangular and one was a "V" shape. This data is presented in Figure 45. The "V" shape reduces leakage by 89%. A reduction of this magnitude in the baseline valve leakage fuel component would amount to a savings of .373 M-lb or 3.3% of the total hydraulic system fuel consumption.



LEAKAGE COMPONENT	RECTANGULAR (CC/M)	V-SHAPE (CC/M)
Annulus	5.4	5.4
Orifice	409.1	41.1
Total	414.5	46.5

Energy Consumption/Aircraft	Baseline	V-Shape
	.42 M-Lb	.047 M-Lb (3.3%)

Figure 45. Valve leakage vs. orifice shape

Energy Consumption

M-Lb Per A/C Life		
Baseline (Fuel)	High Overlap (Fuel)	Savings (Fuel)
.42	.11 (26%)	.31 (74%)

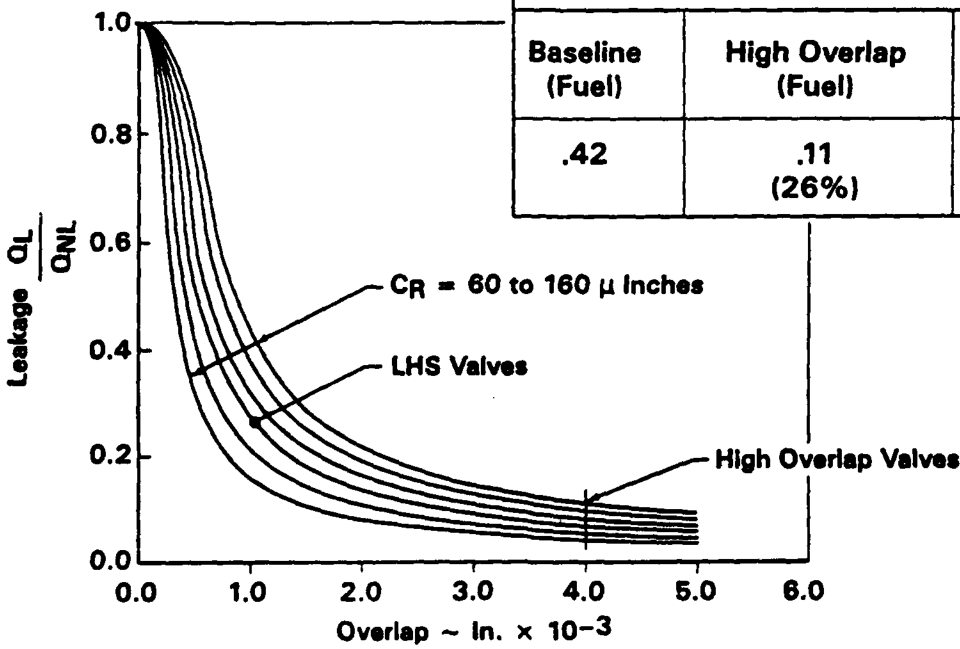


Figure 46. Valve leakage vs. overlap

High Overlap Valves

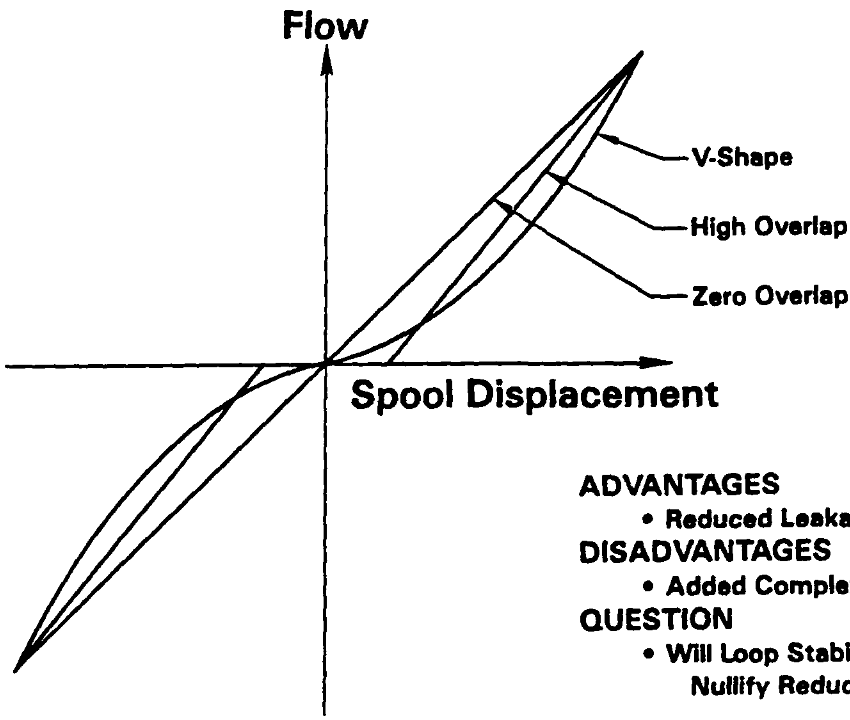
Valve overlap, depicted in Figure 44(c), is commonly used to minimize leakage. Leakage as a function of overlap and clearance is shown in Figure 46. It can be seen that a few thousandths overlap significantly reduces leakage. The valve design used in the LHS test program incorporates about 0.001 inch overlap. If overlap were increased to 0.004 inch, leakage could be reduced 26%. This would decrease the valve leakage fuel consumption component 74% - a savings of 0.31 M-lb of fuel over the aircraft life. This is 2.8% of the hydraulic system fuel consumption.

Non-Linear Valve Operating Characteristics

High overlap and "V" shape orifice valves have non-linear flow characteristics. By substituting the area functions (Area = $F(x_v)$) into the valve flow equation, "V" shape orifices produce parabolic flow characteristics, rectangular orifices produce linear flow characteristics, and rectangular orifices with overlap produce linear flow characteristics with a deadband, Figure 47.

The non-linear characteristics can be compensated for in the electronic amplifiers used to drive the valve torque motor. If amplifier gain was the inverse of the valve gain, the combined gain (flow vs. electrical command) would be linear. For the "V" shape orifice, an infinite gain would be required at null, thus a simple gain schedule would not suffice. A semi-"V" which produces a higher flow gain in the null region (with some additional leakage) would be a good trade-off. An "intelligent" valve amplifier with a micro-processor using digital techniques would provide a better means of compensating for valve non-linearities.

NONLINEAR CONTROL VALVES



ADVANTAGES

- Reduced Leakages (Potential)

DISADVANTAGES

- Added Complexity in Control Loop

QUESTION

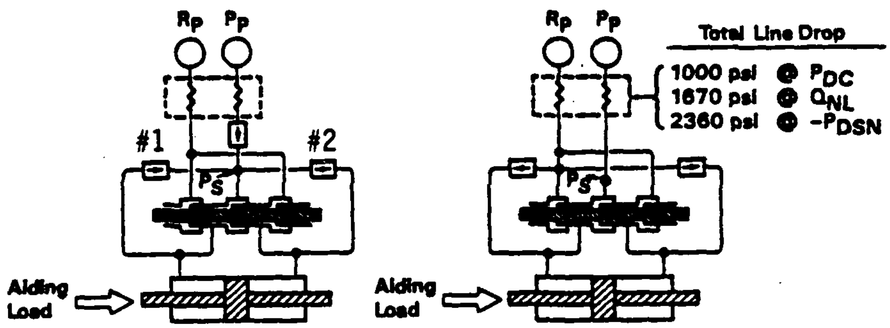
- Will Loop Stabilization and Control Nullify Reduced Leakage?

Figure 47. Nonlinear control valves

Overlap significantly reduces leakage but introduces deadband in the control loop. Dead band is undesirable in high performance flight control servo systems since it produces "back-lash" in closed loop performance and backlash is destabilizing in the outer loop control. Classical design of high performance systems generally demands zero overlap or critical center servo valves to eliminate all deadband. Thus overlap produces conflicting results; on the one hand it reduces leakage and saves energy while on the other hand it degrades dynamic performance. The conventional approach to eliminate deadband is the use of "dither". A dither signal is introduced which causes the spool to oscillate at relatively high frequency across the deadband. The effect of the command signal is then immediate or nearly immediate; the deadband is essentially removed. This technique has been proven effective with compensation, however, valve leakage resulting from the dither may become equal to higher than a valve with zero overlap. Again, an intelligent amplifier, using digital techniques, should provide acceptable performance and still retain the leakage advantage of overlap.

2.4.5.2 Aiding Load Recovery Valves. The conventional servo valve/actuator consumes the same amount of energy regardless of whether the piston load is aiding or opposing. A pump must deliver the same volume of fluid to move the actuator piston an incremental amount regardless of the direction of the load force or even if the load is zero. Energy consumption could be reduced if aiding load forces could be employed effectively.

Aiding load recovery (ALR) concepts are illustrated by the hydraulic circuits in Figure 48. Assume the control valve in Figure 48(a) commands the piston to move toward the right and the load force is pushing toward the right. If the load force is of sufficient magnitude to raise the pressure in the right hand cylinder chamber above the supply pressure, check valve #2 will open. This allows fluid to flow from the RH cylinder chamber to the LH chamber via the check valve and servo valve without drawing fluid from the pump, thereby reducing extracted energy.



(a) (b)

- Actuators Are Sized for 160% × $H_{M(Max)}$
- Aiding Loads Are Never High Enough To Generate Cylinder Pressure Greater Than P_S
- No Energy Saving Potential for Baseline Design Procedure

Figure 48. Aiding load recovery valve concepts

- Assuming Valve Works Perfectly and No Energy Is Extracted for Return Cycle
  - + Limit Saving Potential 0.23 M-Lb
- Realistic Potential 10% to 20%
  - + Break Even Weight 0.38 Lb /Valve
  - + Requires 36 Return Valves
- Disadvantages
  - + Complexity
  - + Reliability
  - + Dynamic Response
  - + Incompatible With Dual Pressure Systems
  - + Maintenance
  - + Cost

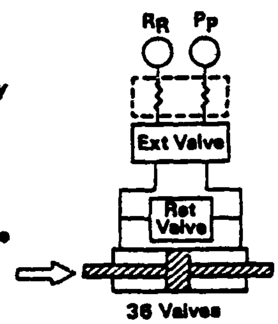
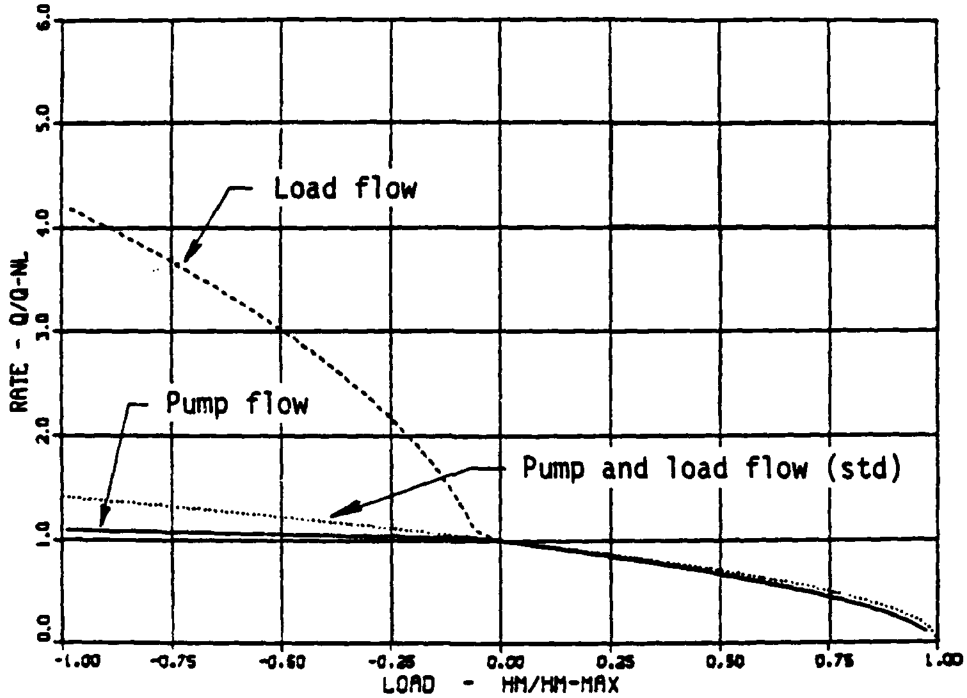


Figure 49. ALR valve energy saving potential

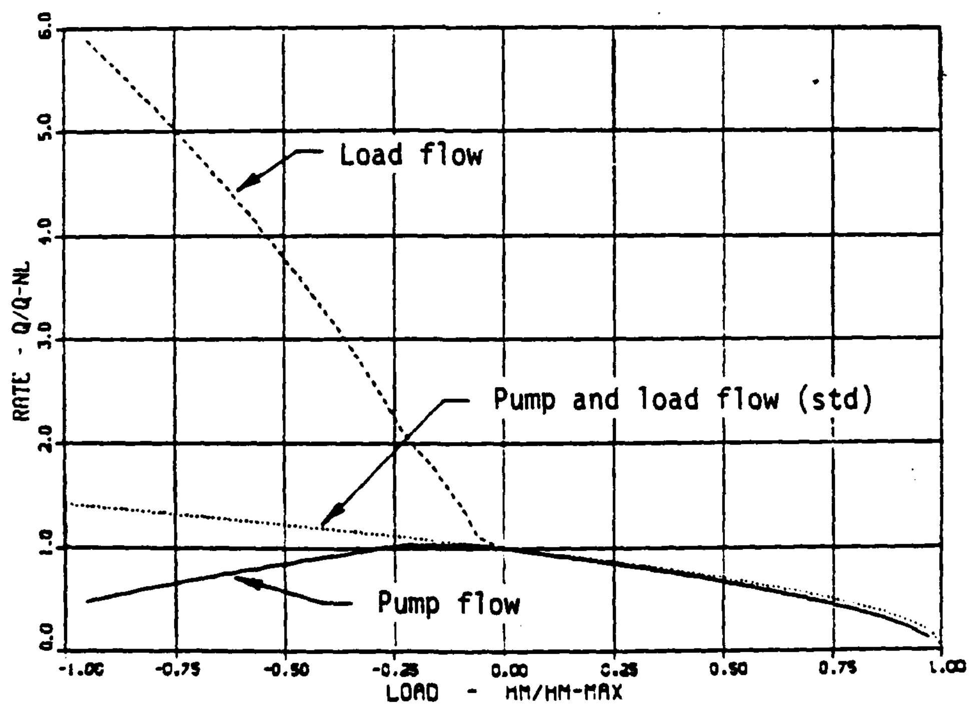
The concept shown in Figure 48 would be impractical for use in the baseline system because the load force must be very high for the cylinder chamber pressure to exceed system supply pressure. Flight control actuators are sized to handle the maximum expected hinge moment (load force). There is rarely, if ever, a flight condition which will "back down" the actuator, and conversely an aiding load which will generate a chamber pressure greater than supply pressure. The baseline F/C actuators were sized for 160 to 200% of maximum hinge moment due to redundancy considerations. An aiding load would have to be more than 160% of the maximum design load to make the cylinder pressure exceed system supply pressure.

The ALR concept can be implemented by resizing the hydraulic lines to induce larger pressure drops and increasing the valve size to maintain the same maximum no-load rate capability. Figure 50(a) and (b) show the theoretical (normalized) results of two preliminary designs. Greater rates under aiding load conditions were achieved with decreased pump flow for circuit (a) and constant pump flow for circuit (b). However, pump size cannot be reduced due to the maximum design load rate requirement. Distribution line weight can only be decreased by a small amount since actuators are supplied from a network and only the final line branch size could be reduced. Valve size must be increased to offset the increase in supply line pressure drop; this increases valve weight and internal leakage.

Applying the aiding load recovery concepts illustrated in Figure 48 to the baseline by re-sizing the supply lines would require increasing the servo valve size by 85%. Leakage losses increase by 0.17 M-lb; usage losses were assumed to decrease by 50% or 0.23 M-lb. Weight losses are unchanged, since the larger valve size plus check valves offset the reduced supply line weight. The net result is a fuel savings of 0.06 M-lb.



(a) Return-to-cylinder flow



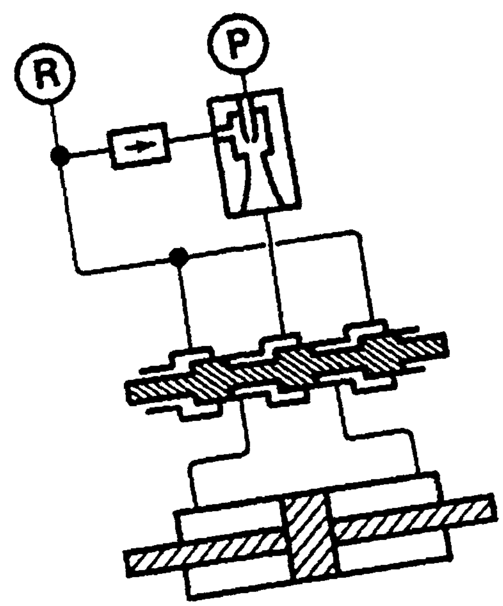
(b) Cylinder-to-pressure flow

Figure 50. ALR valve design results

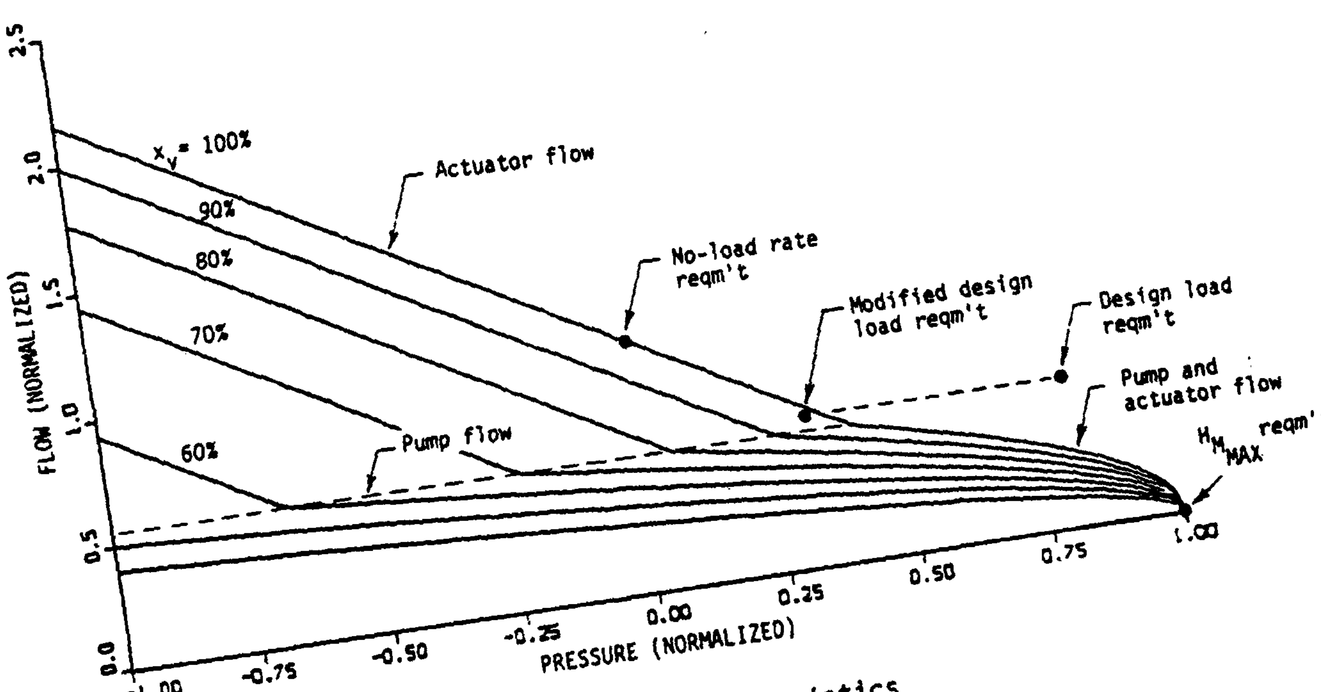
The energy saving potential of the aiding load recovery concept was investigated by applying it to the baseline primary F/C and T/V actuation. An "ideal" extend/retract valve was assumed. (An "ideal"valve saves all aiding load energy.) The usage component of fuel consumption would, therefore, be reduced 50% or 0.23 M-lb. The weight which the "ideal valves" could add to the system and just break even on total fuel consumption is 1.49 lb (each). Since most operation is small amplitude displacement about the null position, a more realistic savings is 10 to 20% of the usage component. This would allow a break even weight of about 0.36 lb per valve. The potential of this concept is summarized in Figure 49.

The aiding load recovery valve does not appear to have much potential for saving energy. If the design requirements could be changed to allow a reduction in maximum opposing load rate requirements, this would permit a decrease in pump size. In this case, the concept would have some potential.

2.4.5.3 Flow Augmentation Control Valves. The Flow Augmentation Control (FAC) valve concept is depicted in Figure 51(a). A jet pump is incorporated upstream of the control valve in conjunction with the aiding load recovery check valves which, under certain operating conditions, pumps fluid from the return line to the valve supply line. This reduces flow drawn from the pump and decreases shaft power extraction. Typical flow augmentation characteristics are shown in Figure 51(b). During high flow conditions, fluid from one side of the actuator is pumped directly to the other side through the flow augmentor and check valve; pressure and return line flows are reduced. The servo valve still controls actuator position. As flow decreases, due to load conditions or valve throttling, jet pumping action decreases until the jet velocity can no longer reduce the nozzle downstream pressure sufficiently below the return line pressure to open the check valve. At flows below this point, the jet pump ceases to function and the valve/actuator operates in the normal manner. As in the ALR concept, the distribution lines are reduced in size to provide large pressure drops and the valve size is increased to maintain the required no-load rate capability.



(a) FAC valve concept



(b) Operating characteristics

Figure 51. <u>Flow augmentation characteristics</u>

The flow augmentation concept was applied to the baseline system and found to provide no potential energy savings due to the baseline design requirements. Potential savings would occur, however, if the design requirements were such that the advantage of flow augmentation could be utilized . The energy saving potential was, therefore, estimated on the basis of "modified" design requirements. The requirements in question are discussed in the following paragraphs along with estimated energy savings.

Conventional Design Requirements. Design requirements typically specified for F/C actuation consist of maximum hinge moment $(H_{M_{MAX}})$, no-load rate $(\dot{\theta}_{NL})$ and a design load point $(H_{M_{DSN}}, \dot{\theta}_{DSN})$. These points are depicted in Figure 52(a). Many designs could satisfy these requirements, however, to minimize weight, the actuator is sized to just meet the $H_{M_{MAX}}$ point (i.e., stall). With the actuator size established, the valve is then sized so that the load-rate capability encompasses both the design load point and the no-load rate point. The load-rate curve for a conventional servo valve/ actuator design is also shown in Figure 52(a).

The design load point for the baseline system is at 80% of $H_{M_{MAX}}$ and 60% of $\dot{\theta}_{NL}$. These values were established by studies conducted at Rockwell on advanced reduced stability aircraft. The design load point, therefore, sizes the control valve, and the no-load rate capability is 34% greater than the no-load rate requirement, Figure 52(a). The FAC valve operating characteristics are shown for these design requirements in Figure 52(b). Flow, plotted on the ordinate axis, is normalized to the no-load rate capability. An alternate scale, on the right, shows flow normalized to the no-load rate requirement $(\dot{\theta}_{NL})$. The design load point flow is at 0.45. Flow augmentation begins operating at 0.55 and provides greater rate capability with less pump flow than the conventional design. However, the additional rate is not required and, if the design requirements are correct, would not be used. Flow augmentation valves under these conditions would provide no real benefit.

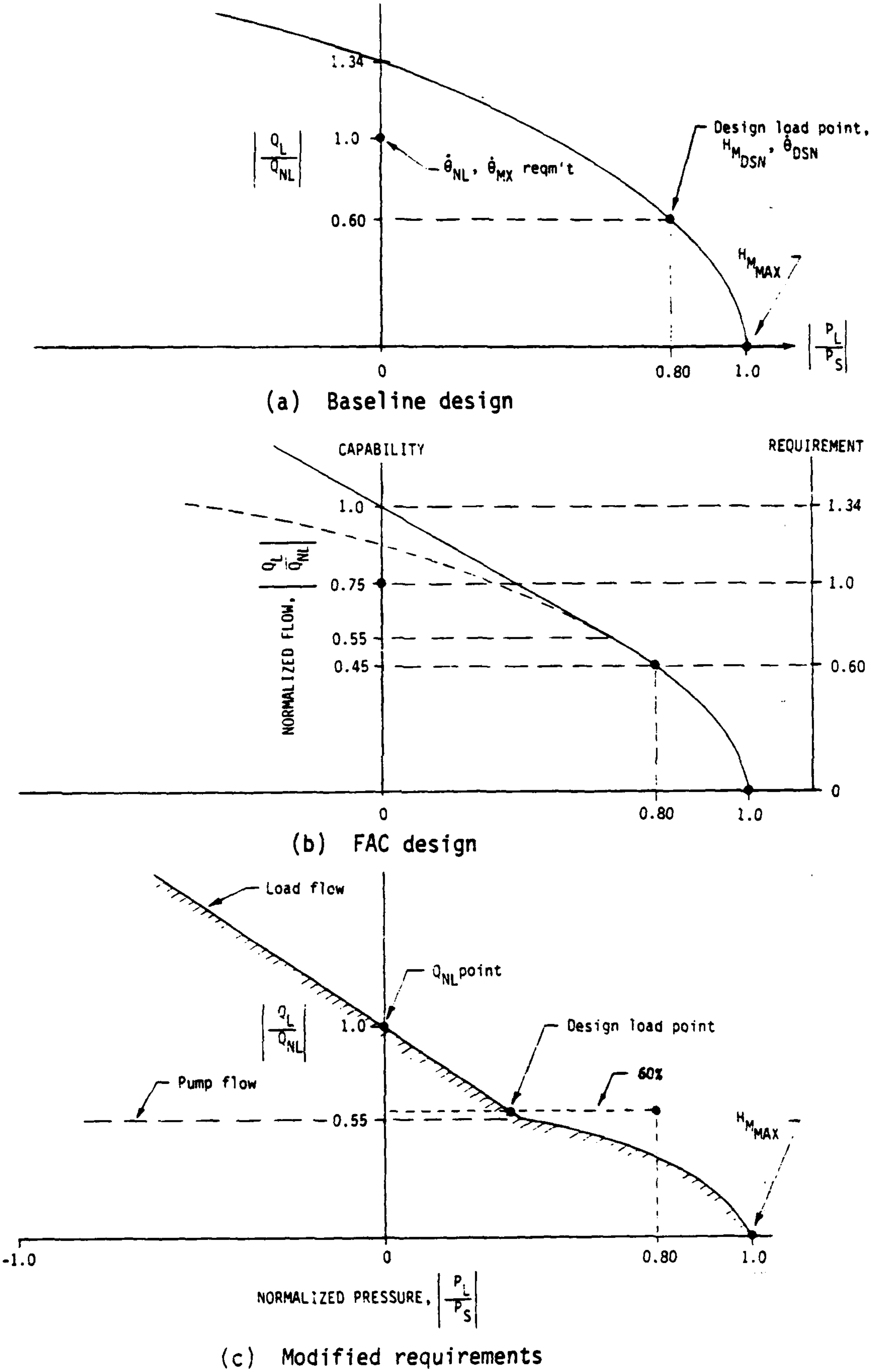


Figure 52. Flow augmentation design points

Modified Design Requirements. If the design load point was on or within the performance boundary (solid curve) shown in Figure 52c, flow augmentation could be employed to save energy. To assess the potential, it was assumed that the design load point was located within the boundary. Estimates were then made for the energy consumption components: usage,leakage and weight.

The configuration studied consisted of the baseline system modified by substituting FAC valves for all servo valves in the F/C and T/V actuation systems (40 dual valves).

The FAC valve characteristics shown in Figure 51(b) were used to estimate the reduction in pump flow and, therefore, work per cycle. These characteristics were based on an extrapolation of data presented in references 8 and 9. Flow depends not only upon amplitude and actuator displacement but also upon cycle frequency. The frequency used is dependent upon the actuator and is given in Table 12. Flow augmentation reduces flow at maximum no-load rate conditions by 45%. Maximum rate is not always required during the cycle. Using the characteristics of Figure 51(b) and the same approach employed in the baseline calculations, a reduction in usage energy consumption of 0.02 M-lb per A/C life was computed.

A weight estimate was made for the FAC configuration. Results are summarized in Table 55. The incremental valve weight increase was conservatively estimated as 0.5 lb/valve; this includes the jet pump, manifold, control valve size increase, and check valve components. The bases for the weight change in pumps, heat exchangers and reservoirs are also listed in Table 55. Weight trade data in Appendix B was used to establish the weight changes. The distribution system was re-sized for a 45% flow reduction and higher branch line pressure drops. A total weight reduction of 25.7 lb was realized by the FAC configuration.

TABLE 55. Flow augmentation weight change estimate

ELEMENT	WEIGHT
VALVES (40 dual x 1.0 lb/valve)	+ 40 lb
PUMP (45% reduction in capacity)	- 15.3 lb
HEAT EXCHANGER (45% flow reduction x 2/3 in lines)	- 6.6 lb
RESERVOIR (29% tube volume reduction, 30% reservoir size for thermal expansion)	- 4.7 lb
TUBING [141.1 lb baseline - 102.0 lb (FAC)]	- 39.1 lb
Total Wt. Reduction	- 25.7 lb

TABLE 56. Flow augmentation energy analysis

o Assume Actuation Requirements Allow Taking Advantage of FAC Valve

ELEMENT	WEIGHT, LB	ENERGY COMPONENTS	FUEL, M-LB
Valves	+ 40	Weight	- .09
Pump	- 15.3	Usage	- .02
Tubing	- 39.1	Leakage	+ .20
System	- 11.3	Pump	- .32
Net	- 25.7	Net	- .22

Pump losses average 14% of rated pump output (see Figure 24). Assuming pump sizes can be reduced by the full amount of the flow augmentation (45%), then the pump loss fuel component would be reduced by 0.31 M-lb. Valve internal leakage was assumed to increase by 45% since larger control valves are required to compensate for the jet pump and associated circuit modifications necessary to make the concept work. This amounts to an increase of 0.2 M-lb of fuel.

The FAC configuration provided a net saving of 0.2 M-lb of fuel per aircraft life compared to the baseline. The incremental changes in the energy consumption components are summarized in Table 56. The additional weight of the valves is more than offset by the decrease in weight of other components. If system design requirements permit, the FAC valve has the potential for reducing energy consumption 2% and decreasing weight 26 lb.

2.4.6 Multi-Pressure Level Systems

One fail operative/two fail safe requirements imposed on current military aircraft necessitate using over-size flight control actuators. Each section of a dual actuator is typically sized to provide full hinge moment in order to meet the specified performance requirements after a single hydraulic system failure. Some of the baseline aircraft flight control actuators are oversized by only 60% due to control effector redundancy; others are oversized by 100%. Tables 20 and 21 list the design factors (hinge moment capability divided by hinge moment required) required for each of the control functions. Figure 12 shows the baseline vehicle control effectors. When both sections of a dual actuator are operating at system pressure, the actuator has excess hinge moment capability. Under this condition, system pressure could be reduced significantly and still meet full actuator hinge moment requirements.

2.4.6.1 Dual Pressure Level Systems. A 4000/8000 psi pressure level system was conceptually designed using a hinge moment design factor of 2.0 for all F/C and T/V actuators. As a first trial, the baseline power supplies and distribution system designs were used, and the total fuel consumption per aircraft was calculated. Fuel usage increased due to larger actuators required for control functions 3 through 10, Table 20. It was thus concluded that increasing actuator size to take advantage of the 4000 psi pressure level is not profitable from an energy standpoint.

A second design approach was investigated: Use the baseline actuators and reduce the supply pressure to 4000 psi during flight modes which do not require full hinge moment capability. Basic design requirements are reviewed in Table 57. From system considerations, it would appear that reduced pressure should only be allowed in mission legs 2, 3 and 6, (reference Table 3) since these legs do not require high hinge moments.

A dual pressure level system was conceptually designed where 4000 psi system pressure is used for mission legs 2, 3 and 6 and 8000 psi for legs 1, 4, 5 and 7. Switching logic for this system is shown in Figure 53. This logic could be mechanized in the F/C computer; some additional sensors/signals would be required. Safety dictates that any failure should cause the system to revert to full pressure. Using the baseline hydraulic system supplies and distribution system, fuel savings were computed for this approach (Design No. 2), and the results are presented in Table 58. A reduction in fuel consumption of 0.44 M-lb per aircraft life was achieved. This amounts to a 4% savings in the total fuel consumption of the baseline.

TABLE 57. Design requirement review for mission leg

*MISSION LEG	REVIEW COMMENTS
1 and 7	o Must use utility actuators. Since it would not be efficient to size these actuators for reduced pressure, system pressure must be 8000 psi.
2, 3 and 6	o Low to moderate hinge moments are required. Eighty percent of max hinge moment is sufficient; system pressure can be 4000 psi.
4 and 5	o The risk associated with these legs is high, therefore, full 8000 psi capability should be provided for survivability/reliability considerations.

*See Table 3

TABLE 58. Dual pressure level fuel consumption

	BASELINE (M-LBS)	DESIGN 1 ALL MODES (M-LBS)	DESIGN 2 MODES 2, 3, AND 6 (M-LBS)
Usage	0.48	0.29	0.30
Valve Leakage	0.23	0.16	0.17
Pump	0.70	0.42	0.47
Weight	9.47	10.32	9.50
Total	10.88	11.18	10.44
Δ	Basis	+.30 (+2.8%)	−.44 (−4.0%)

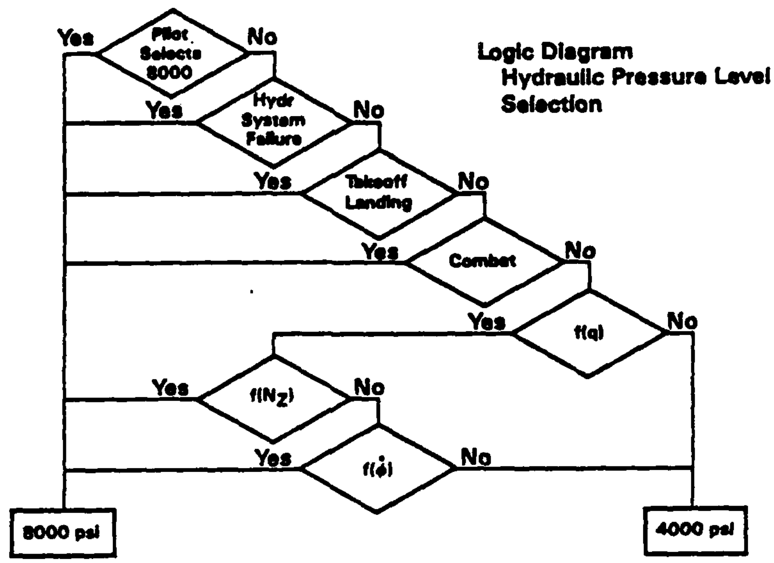


Figure 53. Dual pressure level logic diagram

TABLE 59. Dual pressure level advantages/disadvantages

ADVANTAGES

- **Energy Savings**
- **Increase in MTBF**
- **Reduced Heat Rejection**

DISADVANTAGES

- **Valve Parameter Changes**
  - K_Q
  - K_P
- **System Complexity**
  - **Additional Sensors**
  - **More Complex Loop Control Laws**

In addition to saving energy, lower system pressure offers the advantage of reduced pump leakage, decreased power consumption and less pump wear. Control valve throttling losses are less because of better matching between the actuator output force capability and the load.

The dual pressure concept was initially investigated by Rockwell for the ATF program, and found to have considerable merit for reasons other than energy savings. Some of the projected advantages and disadvantages excerpted from these studies are listed in Table 59. The ATF vehicle was entirely FBW in design. As such, it had the capability to handle valve flow gain changes in actuation control loops.

Control valve characteristics, in particular flow and pressure gain (K_q and K_p), are functions of the supply pressure. If supply pressure is reduced one half, flow gain is reduced 30%. Gains in the F/C computer must, therefore, be increased to maintain comparable loop performance. Valve pressure gain is also reduced which affects stiffness, resolution, and deadband. These changes must be accommodated by F/C computer changes or by accepting a reduction in actuation performance.

Additional sensors are required to provide the information necessary for pressure level selection. These sensors must be failsafe or redundant. The increase in complexity due to pressure level logic, actuation loop gain changes, and sensor redundancy mean more complicated control laws in the F/C computer which increases the computational load and necessitates additional capacity.

2.4.6.2 Multi-Pressure Level Systems. More than two operating pressure levels could be used to further optimize efficiency. An algorithm must be developed which would set system pressure to the minimum necessary for particular flight conditions. The effort required to quantify the advantages of such a system is inordinately large; the multi-pressure level concept was, therefore, not pursued. Practically, it would appear that the bulk of the energy savings is provided by the two level system. Gains for a 3, 4 and 5 level system diminish while complexity increases. The concept could be carried to the limit where system pressure is continuously adjusted to meet load demand. Again, considerable study would be necessary to prove the additional complexity is justified.

2.4.7 Hybrid Electro-Mechanical/Hydraulic System

Utility actuators are generally not well suited for use in 8000 psi hydraulic systems because of their relatively low force output requirements. For example, an 800 lb output actuator has only 1/10 of a square inch of working piston area. Small sizes such as this are not efficient weight-wise. Advances in electro-mechanical (EM) technology during the past decade make EM a viable alternative to hydraulics for many applications, particularly in low usage applications which do not require servo control. EM actuation was investigated to identify possible advantages in energy consumption, Table 60.

To determine the energy savings potential of this approach, utility actuators in the baseline vehicle were replaced with an equivalent set (on the basis of load, stroke and rate) of electro-mechanical actuators. This was done without regard for any special design features of the actuators. Fuel consumption was then computed. Table 61 compares the fuel consumption components with the baseline system. The EM usage component is lower than the baseline, however the magnitude of both values are relatively insignificant compared to primary controls usage. Leakage or quiescent losses for both were assumed zero or negligibly small. The weight fuel consumption component is lower for the EM design due to a minor weight reduction. In summary, the EM utility system approach saves 0.07 M-lb of fuel and reduces weight 18.5 lb. The basis of this estimate is presented in the following paragraphs.

The three components of fuel consumption (usage, quiescent losses, and weight) were computed as follows:

Usage Component. The design load point power is listed in Table 22. This power is multiplied by the cycle time to obtain energy consumption per cycle. Cycle time is, by definition, equal to the time required for the

TABLE 60. EM utility system advantages

REPLACE UTILITY ACTUATORS WITH EM TYPE

ADVANTAGES

- Eliminates Hydraulics in Nose of Aircraft
- More Efficient for Small, Low Duty-Cycle Actuators
- Reduce No. 2 Reservoir and System Size

TABLE 61. EM utility system fuel consumption

TOTAL A/C ENERGY COMPONENTS	FUEL CONSUMPTION PER A/C LIFE IN M-LB	
	BASELINE	EM
• Usage		
Primary	.4781	.4781
Utility	.0003+	.0003−
• Leakage	.2268	.2268
• Pump	.7015	.7015
• Weight	9.4690	9.4040
Total	10.88	10.81

- EM Utility System Saves

0.07 M-Lb Fuel (.6%)
18.6 Lb Weight

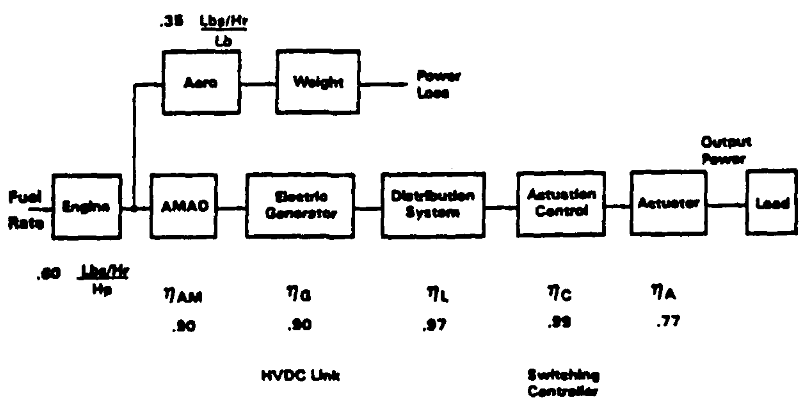


Figure 54. EM efficiency diagram

actuator to move from one extreme to the other and back at the design load rate. The energy thereby used is multiplied by the number of cycles per aircraft life and by the number of actuators per aircraft. The extracted energy is then computed by multiplying by the system efficiencies shown in Figure 54. In mathematical form:

$$P_{DL} = \frac{F(\dot{X})}{6600} \quad hp$$

$$T_{CYC} = 2\,\frac{X_{STROKE}}{\dot{X}_{DL}}\left(\frac{1}{3600}\right) \quad hr$$

$$E_{EXT} = \left(P_{DL} + \frac{30}{746}\right)(T_{CYC})(N_{CYC})(N_{ACT})\,\pi(\eta) \quad hp\text{-}hr$$

Actuation control power losses consist of contactor (relay) coil power and actuator brake coil power. Coil power is about 3 watts for small contactors and 25 watts for large contactors. Typical power for actuator brakes is 25 watts. A value of 30 watts per actuator was used as an average control power loss. Usage fuel consumption was then calculated by multiplying by the fuel consumption coefficient.

Quiescent Component. There is no quiescent loss associated with EM design.

Weight Component. The weight of each EM utility actuator for the baseline system was estimated using the trade data given in Figure B-4 for ballscrew actuators and motors. The actuation controller consists of a 270 VDC contactor; these devices weigh an average 2.5 lb and are nearly independent of the load current. Distribution system weight was determined by establishing an electrical wiring design of equivalent length, number of distribution points, etc. as used in the baseline hydraulic system. The generator weight increment was taken from Figure B-11. This data is a composite estimate based upon IDG used in current aircraft, i.e., the Bendix 270 VDC Link system for the Gulfstream, and other generator data. Hydraulic pump weight was not reduced because the pumps are sized by F/C and T/V requirements. Reservoir weight was reduced by the fluid volume increment required for the utility system.

2.4.8 Advanced Materials

2.4.8.1 Materials Review. New aircraft concepts invariably demand increased speed, range, payload, and performance which in turn require advanced materials with higher strength, stiffness, toughness, and service temperatures. New materials suitable for use in future aircraft are being developed at an accelerating pace, particularly in the areas of composites and powder metallurgy. Current research is the result of intense competition. Companies that provide the highest, strongest, and lowest cost materials will serve not only the aerospace industry, but will also meet new demands in the automotive industry.

A large amount of published information is available covering recent developments in advanced materials. Information presented in this section is the result of a literature search covering composites, powder metallurgy, and supermetals. The applicability of these new materials to hydraulic systems is presented.

2.4.8.1.1 Composites. Composites are a combination of at least two different materials bonded together with an adhesive, and are designed to have properties not possible with any one material acting alone. Composites consist of any combination of fibers, whiskers, and particles in a common matrix and may be classified as:

Fibrous Composite	Fibers in a matrix
Laminated Composite	Fibers in a metal sandwich
Particulate Composite	Particles in a matrix
Hybrid Composite	Several types of reinforcement materials in a common matrix

Outstanding features of composites include high strength-to-weight/ modulus-to-weight ratios, and their ability to be tailored to meet individual load requirements.

Glass fiber/resin composites have been used for thirty years in commercial and aerospace applications. Advanced composites used today in high performance parts are made with fibers such as graphite, boron, or aramid, and matrix materials such as epoxy or PEEK. Advanced composites can have, for their weight, greater tensile strength than aluminum, titanium or steel.

There are, of course, many factors to consider in deciding whether a composite would be suitable for a specific application. For example, fibrous and laminated composites are generally limited to relatively simple two dimensional structural parts with little thickness. Particulate composites can be used for complex three dimensional parts. In any case, components made of advanced composites are very expensive to fabricate compared to parts made with conventional metal alloys.

Reinforcement Materials. High strength, high elastic modulus fibers are the key to producing high performance composites. Graphite is one important material. More than 20 different types of graphite fibers are currently available with strengths ranging from 250,000 to 650,000 psi and elastic moduli ranging from 28 to 75 million psi. Fiber processing determines the tensile strength. Future tensile strengths may reach 800,000 psi. Graphite is available in continuous or chopped fibers pre-impregnated in resin tapes of various sizes or in sheet form. The density of graphite composites is about half that of aluminum and a sixth that of steel.

Aramid fibers were introduced by DuPont in 1971 under the trade name "Kevlar". A notable characteristic of the as-spun fibers is the extraordinary level of crystallinity and orientation which results in tensile strengths five times higher than steel -- on a weight basis. Kevlar is about

40% lighter than fiberglass, 18% lighter than graphite, and has better toughness and ductility than graphite. Both Kevlar and graphite have essentially no dimensional change with temperature.

Metal matrix composites (MMC) are metals reinforced with fibers, whiskers, or particulates. The matrix is generally aluminum, titanium, or steel. Fibers employed include boron, graphite, and tungsten; particulates are mainly silicon carbide. The volume fraction of the reinforcement material varies from 10 to 60% producing strengths from 2 to 10 times that of aluminum and stiffness values of 1/2 to 2 times that of steel. Weight savings, compared to monolithic aluminum alloys, range from 20 to 70%.

The manufacture of metal matrix composites is unconventional. Whisker or particulate reinforced metals are fabricated by mixing reinforcement and matrix powders, cold pressing, followed by hot vacuum pressing. Billets thus formed can be machined, rolled or extruded. Machining takes 2 to 4 times longer than for conventional aluminum because of tool wear, reduced feed rate, and the need for a high surface finish (because of notch sensitivity).

The development of resin composites is about 15 years ahead of metal matrix composites (MMC). MMC's are currently the subject of intense research and many technical problems remain to be solved. Two such problems are the mechanics of fracture and the reinforcement/matrix interface behavior. Current theories of fracture do not apply because of the complex behavior of MMC's during crack propagation. MMC's are costly at present. For example, if hot rolled steel costs a unit price, then monolithic aluminum is 1 to 4 units, SiC/Al is about 600, B/Al is about 1800, and Gr/Al is 4800 to 20,000 units.

Matrix Materials. Two basic types of matrix resins used are thermosets and thermoplastics. Thermosets cure chemically with the application of heat. They have excellent adhesion to reinforcements, superior chemical resistance, and high mechanical properties. Epoxies are a thermoset widely used in advanced composites but they are brittle, have poor damage resistance, and high fabrication costs. Themoplastics cure with the application of heat and can be re-heated and cooled repeatedly (Epoxies cannot). Newly developed thermoplastics such as polyetheretherketone (PEEK) are damage tolerant, do not require refrigeration, and have lower fabrication costs than epoxies. One major advantage is that thermoplastics can be welded by numerous plastic welding processes while thermosets must be mechanically fastened or adhesive bonded.

Composite Properties. Since composites are engineered materials, it is difficult to generalize their physical properties. Parts can be designed to have different properties in different directions. The myriad combinations of reinforcement materials and matrixes obviously affect tensile strengths and elastic moduli. Processing differences can create large variations in properties. Wide differences in longitudinal, transverse, and shear strengths are normal; temperature also affects performance. With this in mind, the following data should be considered as representative and wide variations are possible.

Representative Room Temperature Values

Composite				
Reinforcement	Matrix	Density lb/in^3	Tensile Strength, psi	Tensile Elastic Modulus, psi
Boron	Epoxy	.072	230,000	30,000,000
Kevlar	Epoxy	.050	200,000	11,000,000
Graphite	PEEK	.070	230,000	17,000,000
Boron	Aluminum	.097	216,000	20,000,000
Graphite	Aluminum	.096	150,000	45,000,000

Composite Information Sources. An area located in northern Delaware is a major center of composite research. Principal contributors include the University of Delaware, DuPont, ICI Americas, and Hercules.

The U.S. Air Force has begun work on an integrated composites center aimed at reducing costs and increasing the quality of manufactured parts. The center will be established at McDonnell Douglas in St. Louis, Missouri.

Battelle Columbus Laboratories has an IR&D program to assess the international business climate through 1995 for reinforcing materials as well as a technical evaluation of the field. Products being evaluated include graphite, silicon carbide, aramid, organic fibers, and ceramics.

DOD funding is currently the major driving force in metal matrix composite research. A key source on MMC development is the Metal Matrix Composites Information Analysis Center in Santa Barbara, California.

The single greatest source of information on fibers and resins are meetings staged by the Society for the Advancement of Materials and Process Engineering (SAMPE) headquartered in Covina, California.

Updates on progress in the composites field are covered by many periodicals. Principal magazines include "Metals Engineering", "Ironage", "Machine Design", "Journal of Metals", and "Aerospace Engineering". These periodicals are also excellent sources of information for powder metallurgy and supermetals (see following sections).

Applications. First generation composite actuators are currently being developed by several companies including National Waterlift, HR Textron, and Structural Composites Industries. The state-of-the-art is young and many problems remain to be solved such as porosity, surface finish, and cost. Principal goals are 25% weight savings (over steel), improved fatigue life, and ballistic survivability. Pressure vessels such as accumulators, reservoirs and tubing are also possible candidates for composite fabrication. Tube fittings and valve housings are not likely to be made with composites; other advanced technologies such as powder metallurgy and superalloys are more suitable for these components.

2.4.8.1.2 Powder Metallurgy. Powder metallurgy (PM) is the formation, processing, and consolidation of fine particles to make a solid metal. Two advantages of PM are: 1) the production of alloys with compositions unobtainable by other methods, and 2) the production of finished or nearly finished parts (near net shape manufacturing).

PM products are usually made from commercially available powders that are relatively coarse (particle size larger than 10 microns). The advantages of using finer particles has only recently become known. Very small particles, with diameters less than 10 microns, have unique microstructures and properties unattainable in the larger size ranges. Rapid metal solidification processes are used to make these very fine powders.

Rapid solidification is accomplished by spraying atomized molten metal onto a chilled surface where it is cooled at rates as high as 10^6 °C/sec. Cooling the metal this rapidly makes it possible to retain high temperature crystal structures. Use of rapidly solidified powders results in the final PM product having a uniformly fine microstructure, smaller constituent particle size, and increased alloy strengthening. The net result is improved physical properties.

This new PM technology has opened the door to the manufacture of superalloys. The alloying elements in superalloys tend to segregate during the solidification phase if conventional melt processing is used; this degrades physical properties. Compaction of the same alloys in powder form results in a uniform material which is superior to the wrought metal.

Powder metallurgy processes currently used include:

Forging

Injection molding

Cold isostatic pressing

Hot isostatic pressing

Powder forging employs a powder preform as a billet for forging. The forging operation deforms the blank sufficiently to eliminate porosity and work harden the metal to a degree comparable to conventional forgings.

PM injection molding is similar to the process used for plastic injection molding. The procedure is expensive because it requires the use of metal powders one-tenth the size of conventional powders, and long processing times due to the need to remove a thermoplastic binder. Part sizes are limited to a maximum of 0.25 in. wall thickness.

Cold isostatic pressing is also a slow process but new equipment has recently been developed that shortens cycle times. The principal advantage of this process is the ability to produce intricate, high quality parts.

Hot isostatic pressing (HIP) is the method commonly used to manufacture superalloys for aerospace applications. The process derives its name from the high pressure gas which applies a force uniformly in all directions (isostatically) over the entire surface of the compact in a heated oven. HIP is expensive due to the need for expendable tooling and long cycle times (4 to 16 hours).

PM Alloys. Aluminum PM technology permits the development of new alloy families not possible with ingot metallurgy. Conventional aluminum alloys lose their strength above +300°F. Powder metallurgy offers a means of providing significant strength up to +650°F. The high temperature PM alloys offer such an improvement over aluminum ingot metallurgy that they are competitive with titanium in both airframe components and in high performance gas turbine engines. This is perhaps the most promising area of PM alloy development in the near future. Titanium PM has typically been used to reduce fabrication costs. Recent developments in rapid solidification metallurgy have shown that some of the same advantages obtained in aluminum PM also apply to titanium PM. New titanium alloy families are currently being explored and problem areas addressed.

Metal matrix composites offer extremely fertile ground for future research. One recently developed PM composite, tungsten-carbide grains held together by a cobalt matrix, has revolutionized the tool cutting industry with its high wear resistance. This field obviously overlaps composites discussed in section 2.4.12.1.

PM Properties. Typical properties of aluminum and titanium PM alloys are compared below to conventional ingot alloys.

Typical Room Temperature Values

Alloy	Density, lb/in	Ultimate Strength, psi	Tensile Elastic Modulus, psi
7075 Aluminum Ingot Alloy	.101	72,000	10,300,000
7090 Aluminum PM extrusion	.100	90,000	10,500,000
7090 Matrix 25% SiC Particulate	.102	98,000	17,000,000
Al-Fe-C Aluminum PM Alloy			
Room Temp	0.107	82,000	13,000,000
+300°F		70,000	12,000,000
+450°		61,000	11,500,000
+600°		39,000	11,000,000
6Al - 4V Titanium Ingot Alloy	0.160	142,000	16,000,000
6Al - 4V Titanium PM Alloy	-	133,000	-

Fatigue properties of both aluminum and titanium PM alloys need additional testing to firmly establish design values. It should be noted that none of the high strength alloys discussed have been produced on a large scale; the reproducibility of their properties in production quantities is not clear at this time. Although parts made by powder metallurgy are nearly 100% dense, some microscopic porosity may be present. The effect of this porosity in hydraulic components is not known.

Applications. Although selected PM alloys could be used to replace most metals used in hydraulic systems, tests are needed to demonstrate their suitability. Hydraulic actuators, in particular should be examined, since neither aluminum nor titanium provide good wear surfaces. Hydraulic fittings and valve bodies appear to be excellent candidates for PM technology. Reservoirs and accumulators are probably better suited for filament composites. PM tubing is not currently feasible.

2.4.8.1.3 Superalloys. Recently developed ingot metals with physical properties superior to conventional alloys have been characterized as superalloys. Iron, aluminum, and titanium base superalloys will be discussed.

Iron. Iron-based superalloys contain significant amounts of several alloying elements including cobalt, chromium, nickel, and titanium. The major end-use of these superalloys is in gas turbine engine hot sections which dictate performance requirements of the alloys. Principal applications are in aircraft jet engines and turbine-driven electric generators. The gas turbine vane, blade, and disc materials are superalloys. Although iron-based superalloys could be used in hydraulic systems, lighter weight materials such as metal matrix composites and titanium alloys would provide better strength-to-weight ratios.

Aluminum. Composites have outstanding strength-to-weight ratios and fatigue properties; titanium has excellent high temperature strength and corrosion resistance. The aluminum industry was thus challenged to produce an improved product or face inroads into their share of the aerospace market. One of the aluminum alloys developed to meet this challenge contains the lightest of all metals -- lithium (specific gravity = 0.534). Use of this superalloy saves half the weight that graphite/epoxy composites save, but is one-tenth as expensive. The first Al-Li alloy, 2020, was developed in the 1950's, and was successfully used on the RA-5C Vigilante. Because of low fracture toughness and fabrication difficulties, it was withdrawn from production in 1974. Second generation aluminum-lithium alloys became commercially available in 1986, and currently cost 2 to 3 times more than conventional aluminum alloys.

Al-Li alloys contain 2 to 3 percent lithium; this reduces the density 7 to 10% over current aluminum alloys. When increased strength and stiffness are included, weight savings reach 12 to 15%. Principal use for the new alloys will be in aircraft structural applications; use in aircraft hydraulic systems is not likely (titanium is more suitable). Properties of Al-Li alloys being developed by Alcoa are:

Alloy/ Characteristic	Replacement For	Ultimate Strength, psi	Elastic Modulus, psi	Density lb/in^3
8090A (Damage Tolerant)	2024-T3	65,000	11,400,000	0.092
2090 (High Strength)	7075-T6	86,000	11,400,000	0.094
8192 (Minimum Density)	-	64,000	11,900,000	0.091
8092 (Corrosion Resistant)	7075-T73	71,000	11,700,000	0.093

Titanium. Titanium 6Al-4V is the most widely used titanium alloy in the aerospace industry. It has excellent stiffness, corrosion resistance, and is useful from -320 to +750°F. 6Al-4V is a medium-to-high strength heat treatable alloy that surpasses most steels on a strength-to-weight basis. Drawbacks include relatively high material and manufacturing costs, and susceptibility to galling which limits its usefulness in threaded and sliding contact applications. 6Al-4V is not used for hydraulic tubing because of poor formability.

Titanium 3Al- 2.5V alloy tubing is widely used in commercial and military aircraft 3000 psi hydraulic systems. The advent of 8000 psi hydraulic systems will require thicker wall and/or higher strength tubing. Two characteristics of Ti 3-2.5 tubing make it less desirable for high pressure applications: 1) thicker tube walls will reduce achievable material strength levels; and 2) Ti 3-2.5 tubing cannot be heat treated to improve its strength.

A new titanium alloy, 15V-3Cr-3Sn-3Al, was recently developed with mechanical properties superior to both 6Al-4V and 3Al-2.5V. 15-3 has excellent formability and is heat treatable. 15-3 is particularly well suited for hydraulic tubing. A discussion of this application is presented in the next section.

2.4.8.2 Tubing. Titanium 3Al-2.5V tubing has been widely used in both commercial and military aircraft hydraulic systems for over ten years. This alloy has been broadly accepted because of its high strength-to-weight ratio and reliability. Ti-3Al-2.5V is an alloy which obtains its high strength through work hardening. This is accomplished during the tube drawing process. Difficulty has been experienced in maintaining high strength (125 ksi) properties in thick wall (4000 psi) tubing such as 1-1/4 in x .102. This tubing may have to be designed using a lower ultimate strength (such as 110 ksi) which results in thicker tube walls.

A new titanium alloy, Ti-15V-3Cr-3Sn-3Al (Ti-15-3), is in advanced development for the seamless hydraulic tubing market. Ti-15-3 is substantially stronger and more ductile than Ti-3Al-2.5V. It is a beta alloy which can be age hardened to obtain high strength (181 ksi). Properties of Ti-15-3 are compared with Ti-3Al-2.5V and other tubing materials in Table 62. The strength-to-density ratio of Ti-15-3 is 1.37 times that of Ti-3Al-2.5V at room temperature (1064 vs. 812) and 1.47 times at +450°F.

8000 psi tubing used for this study was designed using both titanium alloys. Design requirements were established in the LHS program and are outlined on Table 63. Tube design is based upon the material ultimate strength at +275°F, burst pressure, and typical manufacturing tolerances. Tables 45 and 46 contain data for pressure and return lines made of the 3Al-2.5V and 15-3 titanium alloys. The weight per foot of tubing filled with fluid per MIL-H-83282 is also given.

TABLE 62. Physical properties of aerospace hydraulic tubing*

TUBE MATERIAL	CONDITION	F_{T_U} ULTIMATE TENSILE STRENGTH (KSI)		F_{T_Y} YIELD TENSILE STRENGTH AT 0.2% OFFSET (KSI)		DENSITY LB/IN^3	F_T U AT ROOM TEMP.	F_T U AT 450°	ELONGATION IN 2 INCHES MINIMUM %
		ROOM TEMP.	+450°F	ROOM TEMP.	+450°F		DENSITY (1000/IN)	DENSITY (1000/IN)	
Ti 15-3	Age Hardened	181	162.5	164	140	.170	1064	956	7
Ti 6Al-4V	Annealed	130	104	120	86.5	.160	812	650	10
Ti 3Al-2.5V	Annealed	85	70	75	56	.162	518	432	15
Ti 3Al-2.5V	Cold Worked	125	101	105	87	.162	773	623	10
Ti Cul. Pure	Grade 2	50	26	40	16.8	.163	306.5	159.5	20
304 CRES	1/8 HD	105	84	75	61	.290	362	290	20
304 CRES	Annealed	75	57	30	22	.290	259	183	35
21-6-9 CRES	Cold Worked	142	116	125	83	.290	490	400	20
AM 350 CRES	Cold Reduced & Tempered	185	165	140	119	.282	656	550	18
6062-T6 Al.	Heat Treat & Aged	42	10	35	8.39	.098	428	111	8
5052-0 Al.	Annealed	26	13	10	7	.097	268	142	-

*Data extracted from References 10 and 11.

TABLE 63. 8000 psi tubing design requirements

	PRESSURE LINES	RETURN LINES
Operating Pressure	8,000 psi	200 psi
Peak Transient Pressure	9,600 psi	4,000 psi
Proof Pressure	16,000 psi	8,000 psi
Burst Pressure	24,000 psi	12,000 psi
Maximum Fluid Temp.	$+275^{0}$	$+275^{0}$
Murphy Prevention	ODD Sizes	EVEN Sizes

The use of reduced wall thickness return line tubing requires a foolproof method of preventing mis-installation of return lines in high pressure locations. A current trend within the industry is to use ODD number sizes for pressure lines and EVEN number sizes for return lines. This arrangement was used in distribution system designs 2 through 5 (see Table 48). The design requirements selected for baseline hydraulic system tubing are summarized in Tables 23 and 24.

The average weight per foot for pressure and return lines made of the two alloys indicates a weight reduction of 22% could be obtained by using Ti-15-3 material instead of Ti-3-2.5. However, since the smaller sizes are more frequently used than the larger sizes, actual weight savings will be less. To obtain a more accurate estimate, the baseline system tubing (Ti-3Al-2.5V) was replaced on a size for size basis. Tubing weight using Ti-15-3 is 303.2 lb compared to 332.5 lb for Ti-3Al-2.5V Tubing -- an 8.8% reduction. This weight reduction translates into an energy savings of 41,000 lb of fuel per aircraft life.

2.4.8.3 Energy Savings. Results of the materials review discussed in section 2.4.8.1 are summarized in Figure 55. Composites show considerable potential for reducing weight, however, the state-of-the-art has not yet matured; full development should be attained by 1995. Powder metallurgy is limited in its application and potential savings. The PM state-of-the-art is not currently at a level necessary for near-term consideration.

TYPE	APPLICATIONS	ADVANTAGES	DISADVANTAGES
Composites (Filament Wound)	— Actuator Cylinders — Reservoirs — Accumulators — Hydraulic Tubing	— Up to 25% Weight Savings Over Steel — Superior Fatigue Properties — Ballistic Survivability	— High Cost — Development Still In Early Stages
Powder Metallurgy (Aluminum and Titanium)	— Valve Housing — Manifolds — Fittings	— Up to 10% Weight Savings Over Steel — Near Net Shape Manufacturing — High Temperature Capabilities	— High Cost — Suitability for Hydraulic Components To Be Proved — Reproducibility of Properties in Production Quantities Is Uncertain
Superalloys (Titanium)	— Hydraulic Tubing — Actuator Bodies	— Up to 50% Weight Savings Over Steel and 25% Over Ti-3AL-2.5V — Excellent Formability — Heat Treatable	—Time of Fitting Attachment vs. Time of Heat Treatment To Be Resolved —Heat Treatment May Cause Tube Bend Warpage

Figure 55. Advanced materials summary

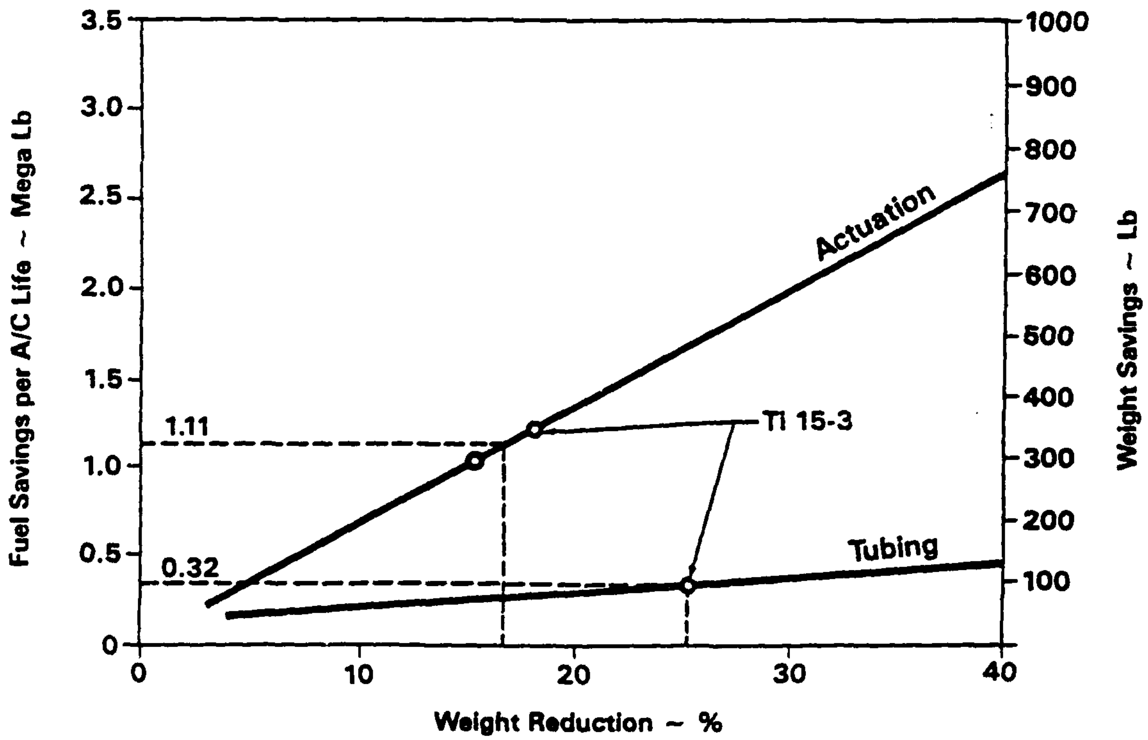


Figure 56. Advanced materials fuel and weight savings

Super alloys have great potential. One improved alloy which currently appears ready for use in aircraft hydraulic systems is Ti-15-3. (See section 2.4.8.1.4.) The energy savings potential of using Ti-15-3 was assessed by comparing it with Ti-2.5V-3Al in the baseline design. A 25% reduction in tubing weight and 15 to 18% reduction in actuator weight was projected. Weight and fuel savings-per-aircraft-life as a function of baseline weight reduction is shown in Figure 56. Assuming a 25% decrease in tubing weight and 16.5% reduction in actuator weight by using Ti-15-3, an energy savings of 1.43 M-lb in fuel and a weight reduction of 408 lb can be achieved. Advanced materials have the greatest potential for saving energy and reducing weight of all the concepts investigated.

2.4.9 Design Margins

Design margins used in aircraft hydraulic systems are more conservative than those used for other areas as shown below.

AREA	DESIGN MARGIN
o Structures	150%
o F/C Mechanical	150%
o Electrical	150%
o Hydraulic Components	
Proof	150%
Burst	200%
o 8000 Psi Hydraulic Tubing	
Proof	200%
Burst	300%

Factors influencing tubing design margins are listed below. Also listed are qualitative comments for each factor which reflect the current state-of-the-art relative to when the margins were originally established.

FACTOR	CURRENT STATE-OF-THE-ART
o Material Property Consistency	Better (Ti-15-3)
o Manufacturing tolerances	Better
o Handling Damage, Scratches, Dents and Etc.	Same
o Actual Surge Pressure	Better test techniques Surges lower in 8000 psi systems
o Bends (ovality)	Autofrettage
o Safety	Hydraulic fuses Reservoir level sensing Multiple controls
o Fitting Stress Concentration	Better fitting designs

Design margins for tubing are significantly higher than any other safety margins used in aircraft design. Burst pressure requirements for 3000 psi tubing and fittings were established over 50 years ago at 400%. It is still the same today even though many advances have been made in fittings, tube manufacturing, materials, inspection, and quality control.

Burst pressure used in the design of 8000 psi tubing is currently 3 x 8000 = 24000 psi. This value is based on a pressure surge allowable of 120% or 9600 psi maximum. If the tubing burst pressure requirement was lowered to 20,000 psi, tubing weight could be reduced. The rationale for lowering the design margin to 20,000 psi involves tubing pressure safety margin, maximum allowable pressure surge, endurance strength, and plastic deformation.

2.4.9.1 Pressure Safety Margin. Safety margins for 3000 psi and 8000 psi tubing are compared below:

3000 psi Tubing (Current)

Burst Pressure - Maximum Allowable Pressure Surge = 12000 - 4050
= 7950 psi

8000 psi Tubing (Revised Margin)

Burst Pressure - Maximum Allowable Pressure Surge = 20000 - 9600
= 10400 psi

Using a burst pressure of 20,000 psi, the pressure safety margin for 8000 psi tubing is approximately 20% greater than for 3000 psi tubing.

2.4.9.2 Pressure Surges. A reduction in the maximum allowable pressure surge from 120% to 115% would permit lowering the burst pressure to 20,000 psi as follows:

$$\frac{\text{Burst Factor - Surge Factor}}{\text{Surge Factor - 100\%}} = 7.5^*$$

*Value for 3000 psi systems, reference 12.

$$\frac{\text{B.F.} - 1.15}{1.15 - 1.00} = 7.5$$

$$\text{B.F.} = 2.275$$

$$\text{Burst Pressure} = 2.275 \times 8000 = 18{,}200 \text{ psi}$$

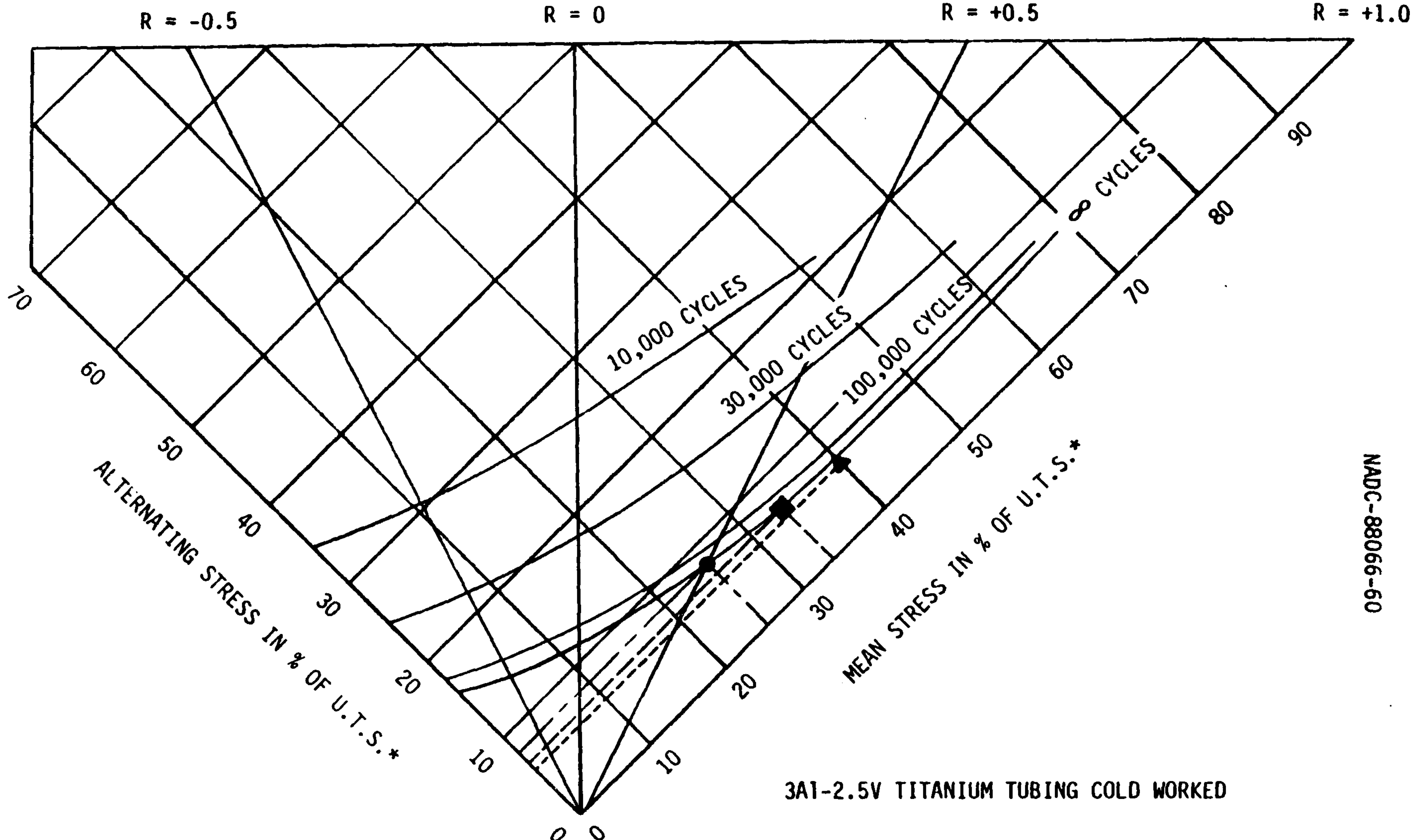
Methods to reduce pressure surges are available and include local fluid velocity control, properly sized restrictors, low surge solenoid valves, and actuator end-of-stroke snubbing. Faster pump response also reduces pressure transients. The amount of pressure surge reduction required -- from 9600 psi to 9200 psi -- is relatively small (4%) and should be achievable with careful hydraulic system design. The allowable overshoot for 8000 psi systems would then be 1200 psi versus 1050 psi for 3000 psi systems.

2.4.9.3 Endurance Strength. The following relationships and a modified Goodman diagram will be used to show that a 20,000 psi burst pressure will not increase stress levels sufficiently to affect tube endurance life.

- Ultimate tensile strength of the tube material is proportional to the burst pressure.
- Mean stress in the tube is proportional to the system operating pressure.
- Alternating stress in the tube is proportional to the pressure surge (overshoot above operating pressure).

	3000 PSI System Design Criteria	8000 PSI System Design Criteria Current	8000 PSI System Design Criteria Revised
Burst Pressure	12,000 psi	24,000 psi	20,000 psi
Maximum Allowable Pressure Surge (ΔP)	1,050 psi	1,600 psi	1,200 psi
Mean Stress	$\frac{3000}{12000} = 25\%$	$\frac{8000}{24000} = 33\%$	$\frac{8000}{20000} = 40\%$
Alternating Stress	$\frac{1050}{12000} = 8.7\%$	$\frac{1600}{24000} = 6.7\%$	$\frac{1200}{20000} = 6\%$

Data for 3000 psi and 8000 psi systems are shown on Figure 57. The alternating stress percentage line for 3000 psi tubing lies exactly on the infinite life curve. The alternating stress percentage lines for 8000 psi tubing lie below the infinite life curve for both the current and revised design margins. This indicates the revised design criteria will not affect tube fatigue life.



○ —— - —— 3000 psi rated system

□ — — — — 8000 psi rated system

△ - - - - - - 8000 psi system (reduced design margin)

*Ultimate tensile strength (F_{tu}) at +300°F

Figure 57. Modified Goodman diagram for internal pressurization cycles

2.4.9.4 Plastic Deformation. When tubing is designed to withstand low-to-moderate pressures and the tube wall is thin, the tensile stress is nearly constant throughout the wall thickness. Under those conditions, stress in the tube wall is related to internal pressure by,

$$S = \frac{Pd}{2t} \qquad \text{Eq. 1}$$

where,
- S = tensile hoop stress
- P = internal pressure
- d = tube I.D.
- t = tube wall thickness

As wall thickness is increased to withstand higher pressures, the distribution of tensile stress across the wall becomes non-uniform and Equation 1 no longer applies. Tubing is considered to be thick wall when the mean radius-to-wall thickness is less than 10 (LHS tubing has a ratio less than 5). Stress in thick wall tubing is usually calculated by,

$$S = \frac{P(D^2 + d^2)}{(D^2 - d^2)} \qquad \text{Eq. 2}$$

where,
- S = tensile hoop stress
- P = internal pressure
- D = tube O.D.
- d = tube I.D.

Equation 2 is based on elastic theory, and produces conservative designs since it does not account for the fact that thick wall tubing has considerable strength beyond the on-set of yielding. Thick wall tubing is more highly stressed at its inner surface than its outer surface. Tubing designed to account for this condition will more efficiently utilize the strength of the material and will have thinner walls than tubes designed using conventional methods. An equation for determining burst pressure with the tube I.D. in the plastic state (beyond the elastic limit) is given below: (Reference 13).

$$P_b = \left(2 - \frac{S_Y}{S_u}\right) \frac{2S_Y}{\sqrt{3}} \ln \frac{D}{d} \qquad \text{Eq. 3}$$

where, P_b = tube burst pressure
S_y = yield strength of tube material
S_u = ultimate strength of tube material
D = tube O.D.
d = tube I.D.

Wall thickness based on 24,000 psi and 20,000 psi burst pressures applied to 0.5 in. O.D. 3Al-2.5V titanium tubing are compared below. Weight savings are also shown.

	Wall Thickness, in.		Weight Savings, %	
	Elastic State Design	Plastic State Design	Elastic State Design	Plastic State Design
24,000 psi Burst Pressure	0.051	0.045	0	10
20,000 psi Burst Pressure	0.043	0.039	13	20

Weight savings obtained for 0.5 in. O.D. tubing would not apply to system savings since a significant percentage of the transmission lines have a minimum wall thickness of 0.020 in. (for handling purposes).

2.4.9.5 Energy Savings. A lower tubing design margin was applied to the baseline. Weight reduction was estimated and fuel savings per aircraft life was computed. A weight reduction of only 3.6 lb was estimated; this provides 0.01 M-lb in fuel savings. This savings is negligibly small and does not justify the reduction in design margins. Results are summarized below:

TUBING DESIGN REQUIREMENTS

CURRENT	3000 PSI SYSTEMS	8000 PSI SYSTEMS
Burst Pressure	12,000 psi	24,000 psi
Allowable Surge	4,050 psi	9,600 psi
Design Margin	4.00	3.00
MODIFIED		
Burst Pressure		20,000 psi
Allowable Surge		9,200 psi
Design Margin		2.50

WEIGHT SAVINGS...........3.60 lb

FUEL SAVINGS.............0.01 M-lb

2.4.10 Thrust Vectoring

The need for short takeoff and landing (STOL) and post stall maneuvering capabilities will likely become requirements for the next generation tactical fighter. Utilization of innovative and conventional thrust vectoring techniques are being examined to enhance the moment balance and control limitation associated with these flight modes. The actuation of thrust vectoring nozzles typically requires large amounts of hydraulic power. This would seemingly preclude the off-loading of control moment generation from aerodynamic surfaces to the T/V nozzles for the sake of improving hydraulic system efficiencies. However, if thrust could be vectored by some other means, hydraulic actuation would not be required. One innovative approach for generating control moments with less control power is the use of hot gas powered thrust diverters. Fluidic control technology has matured to the extent where fluidic devices have been integrated into total systems having capabilities for sensing, stabilization, control, and actuation. This concept is discussed in section 2.4.10.1.

Another possibility is the diversion of hot engine gas to trim nozzles located at the nose and tail of the aircraft. The nozzles direct a small amount of thrust vertically downward at a large moment arm to develop trim moments without using T/V actuators or aerodynamic surfaces. This concept is discussed in section 2.4.10.2.

2.4.10.1 Hot Gas Diverters. Thrust vector control is commonly employed to provide larger magnitude steering moments during low speed flight than can be obtained with aerodynamic surfaces. A number of methods are currently being explored to vector the hot gas of a turbine engine. Actuators can be used to swivel movable nozzles, but this requires elaborately sealed movable joints. Tabs can be inserted to block part of the nozzle exhaust, or vanes can be moved to deflect the exhaust. The injection of a fluid into the nozzle will also vector thrust. The fluid injection technique has the advantage of simplicity and fast response, but typically produces small deflections and exhibits a low primary-flow-to-secondary injection ratio. Other potential problem areas are reduced nozzle and engine efficiency resulting in decreased thrust and increased energy consumption. This is partially offset by less complexity which results in lower weight.

There are many variations of fluidic injection vectoring available. For this study three typical types are examined to show the concepts and define typical weights and energy requirements of fluidic thrust vectoring.

Figure 58 illustrates a deflection nozzle in which hot gas is used to deflect the main flow. Control ports are used on opposite sides to modulate the main flow. At zero deflection the control jet flow is equal and opposite. At full control one port is off and the other full on. The control flow can require up to 10% of the main flow. Flow can be controlled in two axes by adding a second set of control ports. This requires increased control flow.

The bistable control of Figure 59 operates with low pressure bypass air or ram air. The main flow is deflected by the pressure difference between the two control ports. With no control flow, a low pressure area is created which pulls the main jet to the straight wall where the flow is bounded. Deflections up to 30 degrees are possible. While this concept is bistable, it can be made to work proportionally by pulse duration modulation (PDM). Multiple control ports can be used to obtain multi-axis control. Simple electrically operated solenoid valves can be used in this application.

Another control method that does not require high pressure control gas is the overexpanded nozzle shown in Figure 60. If thrust is to be maximized, a nozzle is normally terminated where nozzle pressure matches the ambient pressure. If the nozzle extends beyond this length, flow becomes over-exposed and subambient in pressure. The prevail ng ambient pressure has a boundary layer effect on the ma.n stream. At some point the momentum

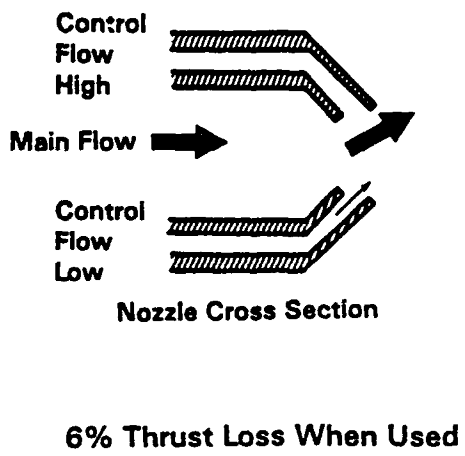


Figure 58. Deflection nozzle

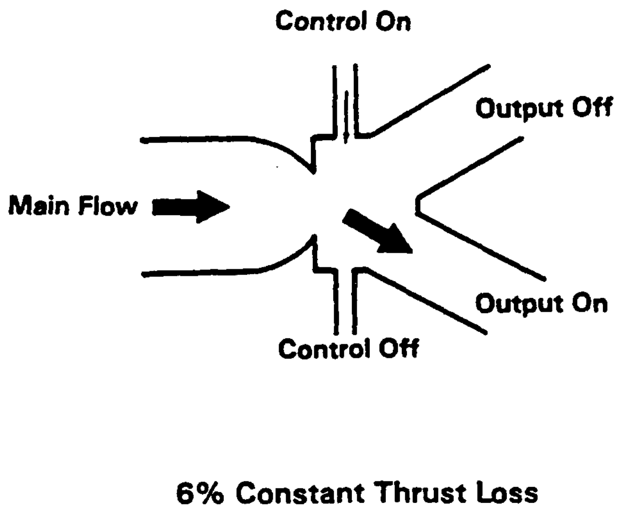


Figure 59. Bistable fluidic control

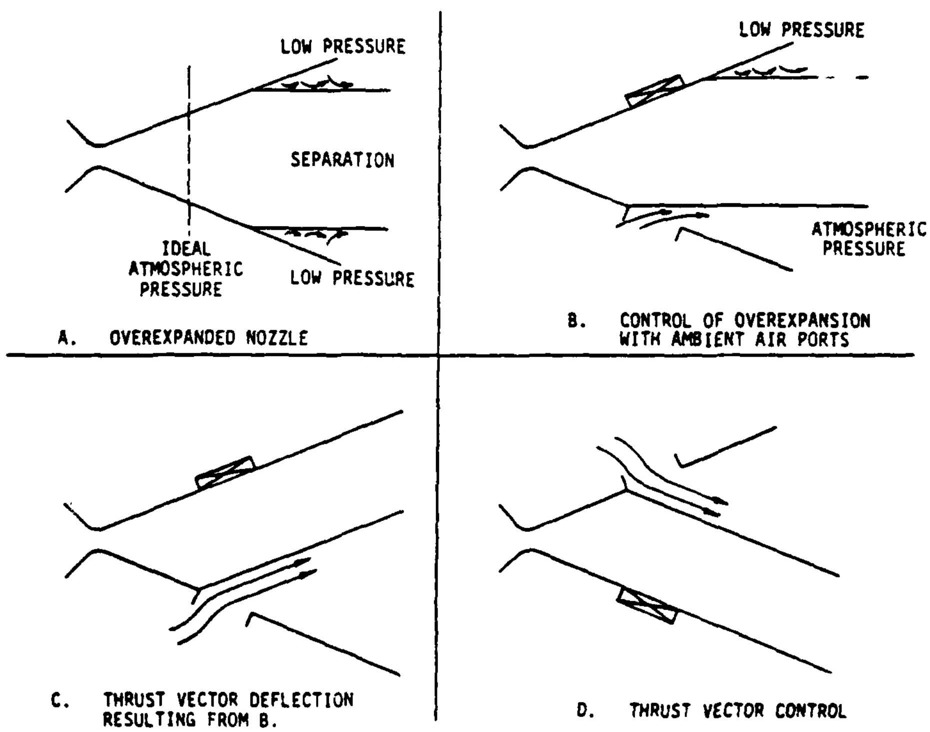


Figure 60. Overexpanded nozzle

in the boundary layer can no longer overcome the higher downstream pressure and the flow separates from the nozzle walls. This free separated flow now entrains ambient air. Since the nozzle extends beyond the separation point the entrainment needs within the nozzle create a counter-flow condition. This lowers the pressure on both sides of the separated flow.

By adding control ports, ambient air can enter through the open control port and partially satisfy the entrainment needs; this causes the local pressure to increase. Since the opposite side port is closed, pressure is lower and flow is deflected. Deflections of up to 15 degrees are possible. To produce axial thrust with no deflection, all control ports are left open to minimize thrust loss due to the ove:expanded nozzle. The overexpanded nozzle has a thrust loss of typically 6 to 8%. The basic advantages are: 1) control ports are not exposed to nozzle hot gas flow; and 2) no external secondary supply subsystem is needed. By installing multiple ports on the nozzle, multi-axis control can be obtained.

The hot gas deflection nozzle was selected for the energy efficiency study because; 1) the overexpanded nozzle is limited to ± 15 degrees of thrust deflection which will not supply the desired amounts of control power; and 2) bistable control requires a series of valves and nozzles which adds to system complexity and weight, and as portions of the thrust are always deflected, there is an inherent loss of thrust even when thrust vectoring is not required.

For the purpose of an energy usage evaluation a design concept and application were developed. The comparisons use deflected angles up to 30 degrees. This is not necessarily a design limit. It was estimated that a maximum thrust loss of 6% would result from thrust vectoring control. While the design concept requires approximately 10% of the main stream flow for deflection, this energy is not all lost as it re-enters the main stream at an angle. There are additional losses in the ducting, control and the re-mixing of th. hot gases which make the 6% loss a reasonable estimate.

Total fuel consumed by the baseline hydraulic system during the 2.7 hour mission is 19,400 lb. Fuel required for the T/V actuators (8) is 497 lb. The hot gas deflection nozzle concept was estimated to use 6% of the thrust for control, or 1164 lb of fuel per mission. This is 667 lb greater than the baseline. The deflection nozzle approach has continuous "direct" fuel consumption whether thrust is being deflected or not, whereas in the baseline hydraulic system, fuel is consumed primarily by actuator movement and secondarily by leakage. Indirect fuel consumption by the hot gas system is less than the baseline because of a projected weight savings. The weight reduction necessary to save 667 lb of fuel/flight is:

$$WT = \frac{667}{2.7 \times .14 \times 2.5} = 705.8 \text{ lb}$$

where, 2.7 = mission time

.14 = fuel consumption rate per pound coefficient

2.5 = weight growth factor

The total equipment weight of the baseline T/V actuation is:

Actuation	257.5 lb
Hydraulic System	84.7 lb
Total	342.2 lb

The equipment weight estimate for the hot gas thrust vectoring concept is shown in Table 64. Since 166.2 lb (baseline T/V weight - hot gas control weight) is less than the required 705.8 lb weight reduction to break even, a sizable energy loss will result from using hot gas thrust vectoring. However, if the main purpose of thrust vectoring is to provide control at low speeds, it would be reasonable to assume that thrust vectoring will only be used for landing, takeoff and combat. It can also be assumed that the thrust vectoring ports can be turned off with no thrust losses in the off position.

TABLE 64. Estimated weight, hot gas control

4 Ducts	4 lb ea.	16 lb
4 Control Valves (Modulate)	8 lb ea.	32 lb
4 Control Nozzles (On-Off)	6 lb ea.	24 lb
4 Misc Hardware	4 lb (lot)	16 lb
Total per Engine		88 lb
Total per Aircraft		176 lb

TABLE 65. Hot gas T/V, fuel consumption

Fuel Consumption (Full Time T/V)		
Elimination of T/V Actuation		-1.09 M-lb
Hot Gas System Weight		+0.62 M-lb
Thrust Loss (6%)		+4.31 M-lb
	NET	+3.84 M-lb
Fuel Consumption (Part Time T/V*)		
Elimination of T/V Actuation		-1.09 M-lb
Hot Gas System Weight		+0.62 M-lb
Thrust Loss Reduced to		+0.69 M-lb
	NET	+0.22 M-lb

*T/V used in takeoff, landing and combat phases.
Assumes no loss when turned off.

Total fuel consumption was computed for both full time T/V and part time T/V (take-off, landing and combat modes). Table 65 shows the difference between these operating modes and the baseline. The full time mode increases total fuel consumption by 3.84 M-lb; the part time concept increases that consumption by 0.22 M-lb.

This cursory look at the use of hot gas for thrust vectoring does not indicate an energy savings. A detailed study would be required, including redesign of the engines, to determine if potential energy benefits exist. The most apparent advantage of some form of fluidic thrust vectoring is the potential for continued control of the aircraft after the loss of all hydraulic power. The application of fluidics or hot gas control for thrust vectoring is still in the early stages of development. While this study did not show an energy saving, the continued development of this technology may provide significant benefits including energy savings.

2.4.10.2 Trim Thrust Vectoring. A trim thrust vectoring concept is illustrated on Figure 61. Engine bleed air is diverted and ducted to the front and rear of the vehicle where it is exhausted through small nozzles orientated along vertical axes. The pitch trim moment is controlled by varying the ratio of air diverted to fore and aft nozzles. A brief review of the concept is presented in Figure 62. Although some benefits may be derived from this concept, its impact on the hydraulic system is negligible; all the baseline actuation systems and control surfaces are still required. Since trim thrust vectoring has negligible impact upon the hydraulic system, and since an aero/propulsion study would be required to ascertain the potential benefits -- which is beyond the scope of effort of the current contract -- investigation of the concept was discontinued.

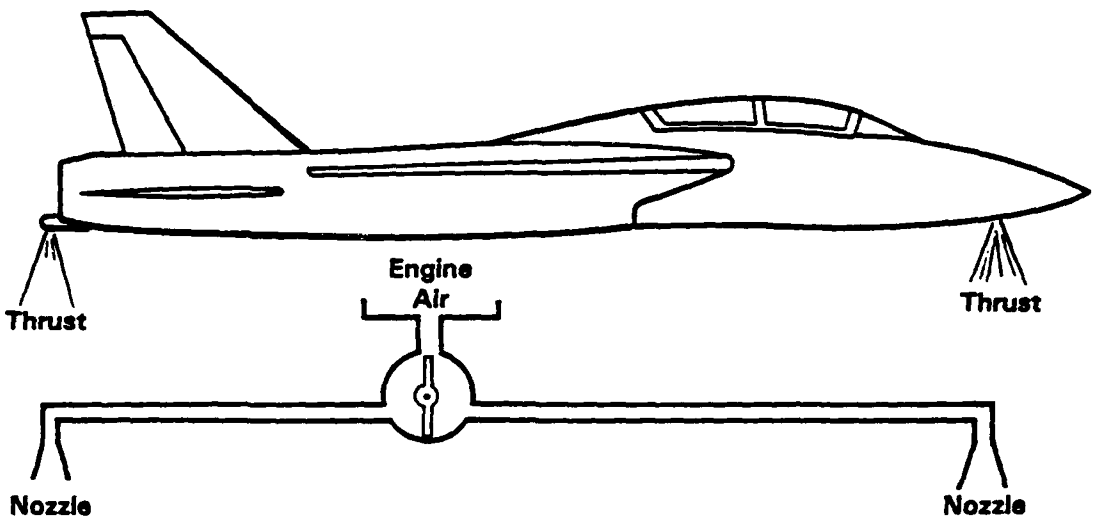


Figure 61. Trim thrust vectoring concept

OBSERVATIONS

- **Long Moment Arm Produce Large Moments With Small Thrust**
- **Eliminates Surface Trim Drag**
- **Eliminates Down-Loaded Surface Equivalent to Weight Savings**

NEGLIGIBLE IMPACT UPON HYDRAULIC SYSTEM

- **Still Need Same Control Surface**
- **Still Need Thrust Vectoring Control**

CONCLUSIONS

- **No Significant Impact Upon Hydraulic Energy Consumption**
- **Requires Aero/Propulsion Study To Assess Advantages**

Figure 62. Trim thrust vectoring review

2.4.11 Vehicle Control Systems

Control system design has a major influence upon power and energy requirements. It brings together advanced technology concepts in control configured vehicles such as optimization of surface commands and adaptive gain capabilities of microprocessor based control systems, direct-drive actuation, alternate control moment generation, and pulse modulated valves. Two control concepts were investigated to determine their potential for energy savings: 1) variable gain/bandwidth and 2) command/control optimization. These concepts are discussed in the following subsections.

A five degree digital computer simulation, illustrated in Figure 63, was used in the investigation. A three axis flight control system, flight control actuation, and atmospheric turbulence were modeled. Baseline gains and bandwidths were established to provide MIL-F-8785C performance.

2.4.11.1 Variable Gain/Bandwidth. Variable gain control systems is an energy savings concept based on the fact that a significant part of an aircraft's total mission time consists of non-critical, low rate maneuvering flight. The versatility and power of the microprocessor can be effectively used to alter system gain and bandwidth to match the lower control system demand while maintaining Level 1 flying qualities. The ultimate effect is to minimize excessive control surface rate demands, overshoots, and reversals that would otherwise occur with the higher gain required for critical, maneuvering flight. The change in gain/bandwidth sensitivity may be a continuous or discrete function of control force, aircraft angular rates, and/or accelerations. The study explored the application of this concept to an advanced multi-mission baseline vehicle having three-axis stability augmentation and autopilot functions. The energy saving benefits derived from demand induced gain/bandwidth variations were parametrically evaluated.

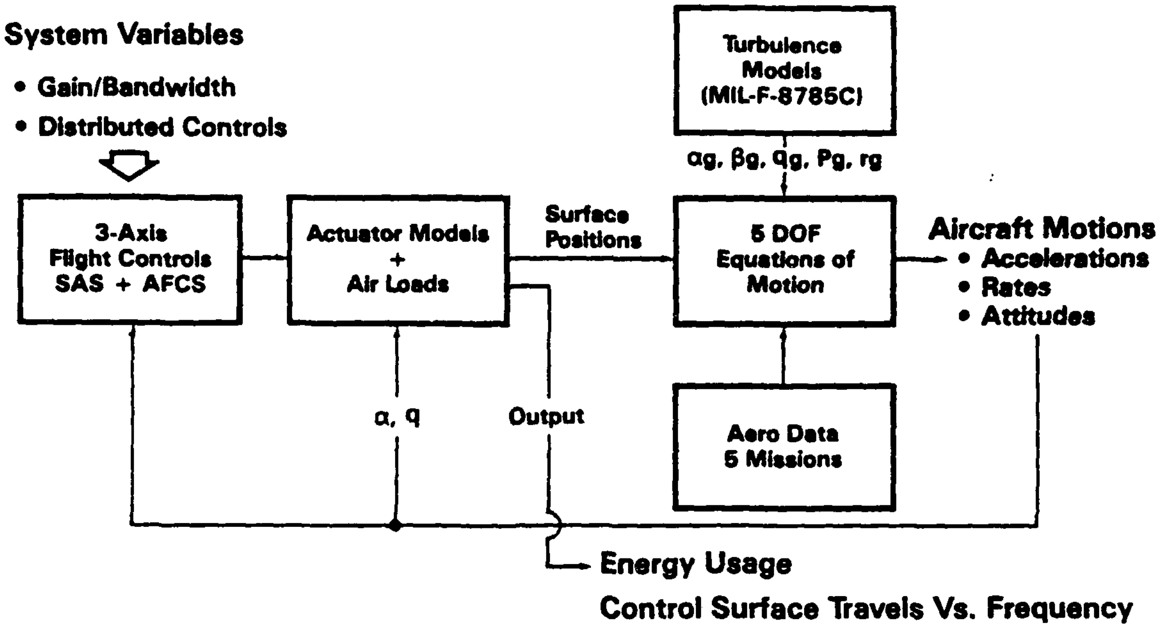


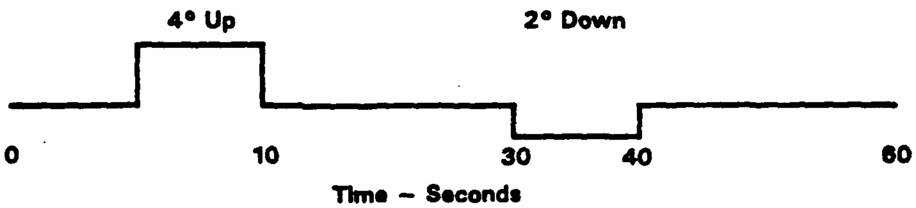
Figure 63. Energy efficient aircraft simulation

CONTROL SYSTEM CONFIGURATION:

- Pitch Attitude Command/Attitude Hold
- Pitch Rate Damping

SYSTEM DISTURBANCES

- Discrete Attitude Commands; Two per Minute



- Angle-of-Attack and Pitch Rate Gusts per Turbulence Models of MIL-F-8785C Flying Qualities Specification

Figure 64. Variable gain approach, disturbances

The pitch axis control system was investigated for two flight conditions: 0.8 M_n at 40,000 feet and 0.27 M_n at sea level. Disturbances to the system consisted of discrete attitude commands and gust inputs, Figure 64. The horizontal surface actuator average displacement as a function of frequence was determined by sampling surface position periodically . Data were collected for nominal, twice (+6db) and half (-6db) control system gains, Figure 65. The energy consumed by the horizontal surface actuators was also computed for each flight condition and gain, Figure 66. Energy consumption for the cruise flight mode is strongly dependent upon control system gain. Energy consumption in the landing mode increases with gain but at a lower dependency.

A variable gain/bandwidth control system was conceived wherein the gains were reduced by 6db in the cruise and loiter mission modes only. This was done because the savings potential in the take-off and landing modes is minimal and maximum performance is needed principally in the dash and combat mission legs. Based upon the data shown in Figure 66, and the methods of computing fuel consumption developed in Section 2.1, fuel consumption would be reduced by 0.21 M-lb over the life of the aircraft. These energy savings can be accomplished with minimal impact upon hardware and consequently, weight.

Another advantage of the variable gain/bandwidth concept is a reduction in actuator cycles which in turn reduces maintenance and improves actuator life. It is recommended that energy consumption and actuator life be a consideration in the flight control law development for new advanced aircraft. Results of this investigation are summarized in Figure 67.

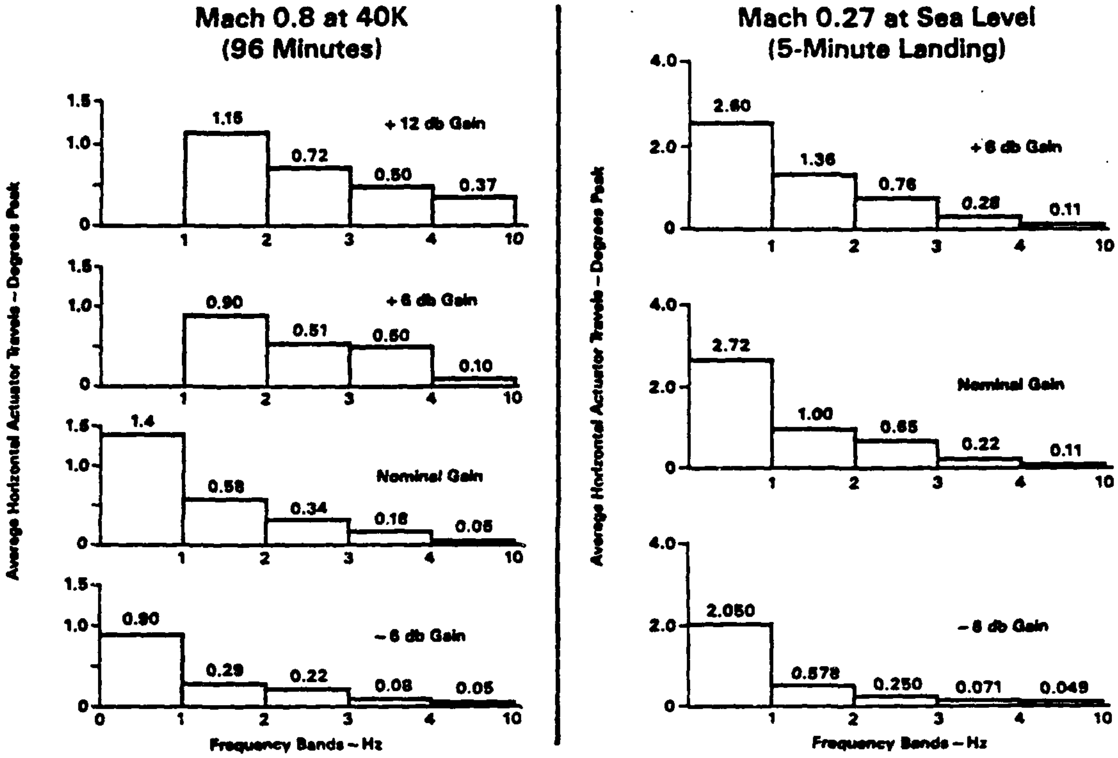


Figure 65. Variable gain approach, gain levels

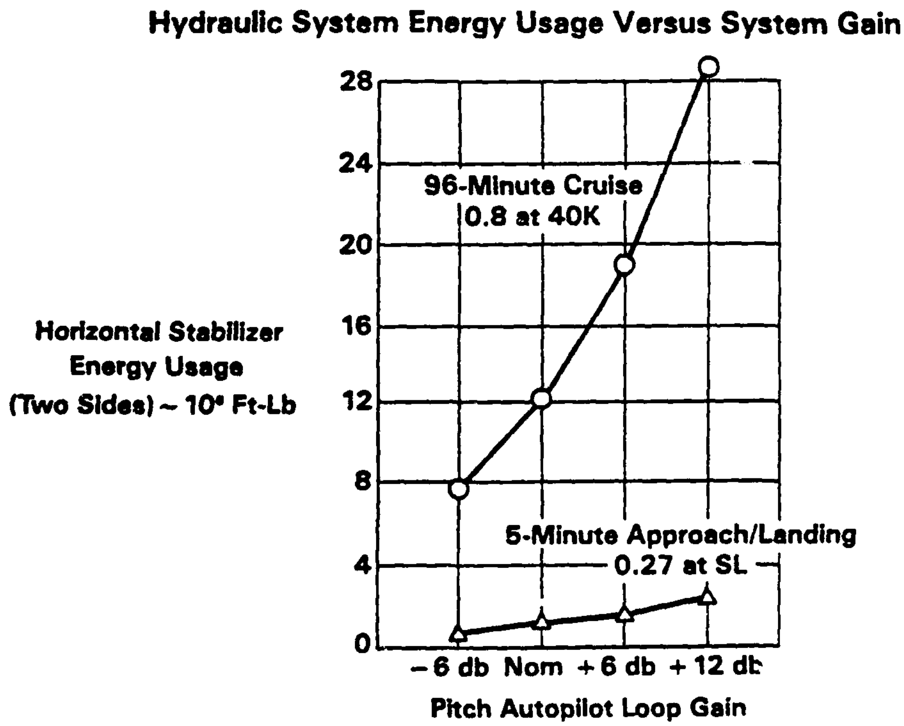


Figure 66. Variable gain approach, energy consumption

ENERGY SAVINGS **.21 M-LB (1.9%)**

(Applying Savings to F/C and T/V Actuators in Cruise and Loiter Modes)

CONCLUSIONS

Energy Savings Can Be Obtained With Minimal Impact on Hardware

Flight Control Law Development Should Include Consideration of Energy and Actuator Life

Figure 67. Variable gain/bandwidth, conclusions

2.4.11.2 Command/Control Optimization. Movement of control surfaces extracts energy from the hydraulic system regardless of the aerodynamic effect upon the aircraft. The effectiveness of control surfaces varies with airspeed, altitude, and vehicle configuration. For instance, outboard ailerons are generally more effective than inboard ailerons due to the roll moment arm (distance of the surface from the centerline of the aircraft). It may be better to utilize only the outboard ailerons until reaching their efficiency limit before deflecting inboard ailerons. Further, it may be more efficient to increase the number of ailerons so that only the most effective areas are utilized. This would also provide advantages in survivability. Some control surfaces, such as leading edge flaps in flexible wing aircraft, become totally ineffective in certain portions of the flight regime and could be shut off to conserve energy. Other controls even become counter-effective, that is, produce the opposite effect desired in specific flight conditions. The classic example of this control reversal is at high "q". The use of thrust vectoring provides additional options for flight control optimization. In some portions of tne flight envelope control by thrust vectoring is more effective and perhaps more efficient than control by aerodynamic control surfaces.

With the computational capability now available for fly-by-wire control systems, it is possible to optimize control moment generation for efficiency. As part of this study effort, the energy savings potential for command and control optimization was evaluated from an overall standpoint. Even though a concept saves hydraulic energy, it would not be energy efficient if it increased aerodynamic drag or introduced undesirable control cross-coupling effects that could produce energy losses.

In general, control surface effectiveness increases with airspeed (dynamic pressure), i.e. smaller surface deflections are required to produce the necessary moment. Since the control surface moves a smaller distance in a given time period, the surface actuator flow rate is lower. This relationship is illustrated in Figure 68. Thrust vectoring effectiveness decreases with increasing dynamic pressure -- the opposite of control surface effectiveness. A control system which optimizes control at each dynamic pressure operating point will be the most energy efficient and will minimize hydraulic flow requirements.

The command optimization concept is depicted in Figure 69. Hydraulic flow requirements for combined control, that is thrust vectoring and aerodynamic surfaces, is reduced from the requirements of aerodynamic control alone. Reducing peak flow requirements is essential to minimizing the size and weight of the hydraulic system (see Section 2.3.2). The effects of combined control were considered in developing the baseline concept.

It became apparent during the study that command optimization would save energy. However, command optimization is more an effect of aerodynamic control system design than hydraulic system design. Hydraulic requirements must be evaluated during the command optimization process, with special emphasis on reducing peak flow requirements. Hydraulic considerations are summarized in Figure 70.

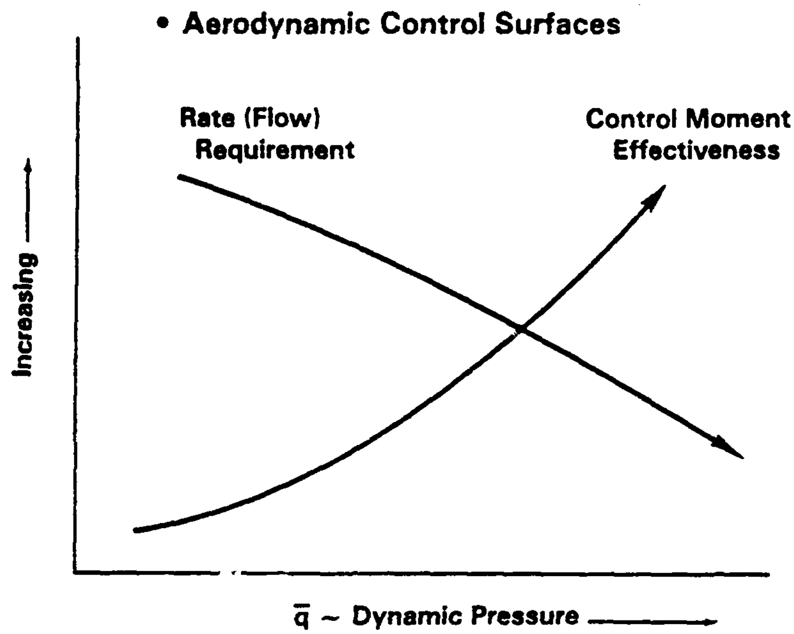


Figure 68. Command optimization, flow requirements

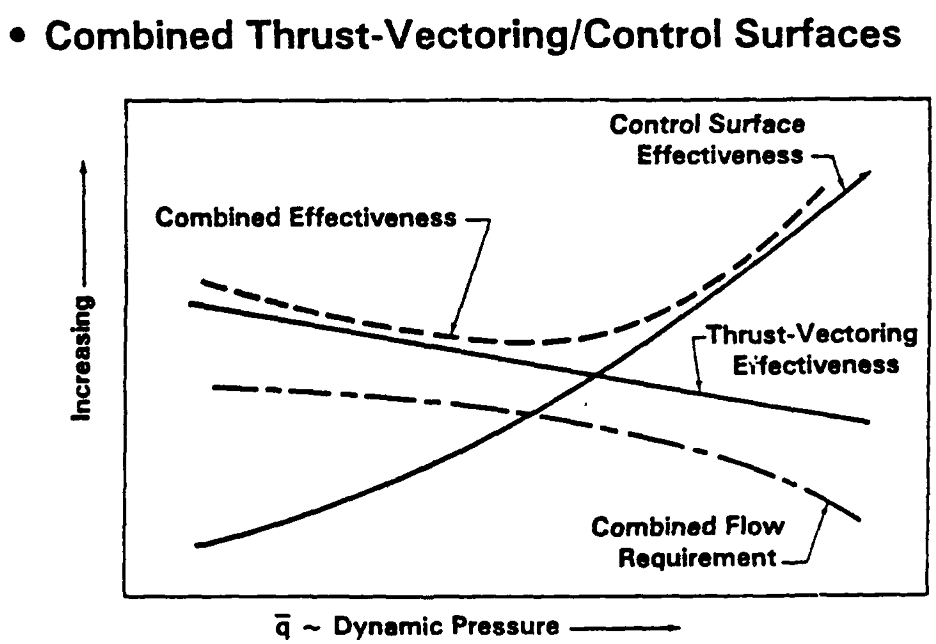


Figure 69. Command optimization concept

HYDRAULIC SYSTEM DESIGN FOR COMMAND OPTIMIZATION

- Hydraulic System Design Requirements Must Influence Control Law Development
- Hydraulic Designer Must Develop Trade Data To Demonstrate Effects of Requirements
- Increase of 1 GPM Is Equivalent to 7.6 Pounds of Fuel Per Flight for Baseline Aircraft

Figure 70. Command optimization, hydraulic considerations

COMMAND OPTIMIZATION

CONTROL EFFECTORS

PITCH	ROLL	YAW
• Horizontals	• Outboard TE	• Rudders
• Inboard TE	• Horizontals	• Vectored Thrust
• Vectored Thrust	• Vectored Thrust	

- Effectors Sized for Worse Case
- System Sized for Control Requirements
- Reconfigurable Controls Used To Select Required Combination of Effectors
- Control Law Development Must Consider Impact Upon Hydraulic System Energy Consumption

Figure 71. Command optimization, control effectors

The control effectors and system considerations for command optimization are listed in Figure 71. The control effectors must be sized for "worst case" which means command optimization will not reduce actuator size; pump and line size may be reduced. Command optimization will reduce energy usage by reducing control commands.

The investigation shows that command optimization mainly affects aerodynamic and flight controls with very little affect on hydraulic design. Command optimization will not result in new design techniques or concepts for hydraulic systems. Command optimization is basically an aerodynamic and control concept for saving energy and not a hydraulic concept. Therefore, it was not pursued in greater detail.

2.5 COMPARATIVE ANALYSIS

The energy saving concepts investigated were rated comparatively. As discussed in Section 2.1, total fuel consumption over the life of the aircraft was selected as the common basis for comparing all concepts. The difference between fuel consumed in the baseline system and fuel consumed when applying the concept to the baseline was computed and referred to as energy savings. This data is summarized in column two of Table 66. Qualitative ratings were established by the procedure discussed in Section 2.1.5; this data is contained in columns 3 through 7 of Table 66. A figure of merit (FOM) was computed for each concept by the procedure outlined in Section 2.1.5.; this data is presented in column 8 of Table 66.

FOM is a parameter by which the concepts were ranked in order of their energy saving potential. Those concepts which had negative savings (actually consume more energy than the baseline) were removed from consideration. The remaining concepts are listed in Table 67. Flow augmentation had zero energy saving potential when applied to the baseline. When applied to the target system it was found to have good potential and was therefore included in the listing.

TABLE 66. Concept ratings

CANDIDATE CONCEPTS	ENERGY SAVINGS, M-LB	AVERAGE RATINGS					FOM RATING
		R&M	LCC	DEVELOPMENT RISK	PERFORMANCE	SAFETY	
PUMPS	+.45	+.01	-.25	-1.43	0	0	.37
IAPS	-1.45	-.76	-.75	-.28	0	+.29	---
DISTRIBUTION SYSTEM	+.35	+.01	0	-.43	0	+.03	.34
ACCUMULATORS	+.12	+1.46	+1.0	-.01	+.05	0	.14
ROTARY VANE	BASIS	+1.25	+1.0	-.37	+.63	0	---
VARIABLE DISPLACEMENT	-.95	-.9	-1.0	-.70	-.25	-.64	---
SLIMLINE - (POD)	-.45	-.45	-.82	+.37	-.55	0	---
SLIMLINE - (HINGLELINE)	-.08	-1.07	-.90	+.20	-.88	0	---
PRESSURE INTENSIFIERS	-1.17	-.9	-.9	-.63	-.38	-.75	---
AIDING LOAD RECOVERY	0	-.44	-.20	-.77	0	-.30	---
FLOW AUGMENT	0/+.5	-.28	-.20	-.45	+.11	-.22	0/.47
NONLINEAR VALVES	+.37	-.14	-.03	-.28	0	0	.36
DUAL PRESSURE SYSTEM	+.44	0	-.52	-.50	0	-.11	.41
HYBRID HYD/EM	+.07	+.15	+.18	-.15	0	+.02	.07
ADVANCED MATERIALS	+1.43	-.22	-.75	-.75	+.05	-.10	1.30
DESIGN MARGINS	0	-.04	-.03	-.6	+.03	-.06	---
TRIM THRUST VECTORING	?	-.28	-.72	-1.10	+.11	-.44	?
HOT GAS THRUST VECTORING	-.22	-.12	-.17	-.97	-.12	+.09	---
VARIABLE GAIN/BANDWIDTH	.21	-.62	-.41	-.15	-.55	+.01	.19
COMMAND OPTIMIZATION	?	-.08	-.37	-.67	-.73	0	?

TABLE 67. Concept rating summary

CANDIDATE CONCEPTS	ENERGY SAVING M-LBS	AVERAGE RATINGS					FOM RATING	HARDWARE DEMO
		R & M	LCC	DEVELOPMENT RISK	PERFORMANCE	SAFETY		
Advanced Materials	+1.43	-0.22	-0.75	-0.75	+0.05	-0.10	1.30	
Dual-Pressure System	+0.44	0	-0.52	-0.50	0	-0.11	0.41	√
Pumps	+0.45	+0.01	-0.25	-1.43	0	0	0.37	√
Nonlinear Valves	+0.37	-0.14	-0.03	-0.28	0	0	0.36	√
Distribution System	+0.35	+0.01	0	-0.43	0	+0.03	0.34	
Variable Gain/Bandwidth	+0.21	-0.62	-0.41	-0.15	-0.55	+0.01	0.19	
Accumulators	+0.12	+1.46	+1.00	-0.01	+0.05	0	0.14	
Hybrid Hyd/Em	+0.07	+0.15	+0.18	-0.15	0	+0.02	0.07	
Flow Augment	0/+0.50	-0.28	-0.20	-0.45	+0.11	-0.22	0/0.47	

2.6 ENERGY EFFICIENT TARGET SYSTEM

An energy efficient system, entitled Target System, was created by incorporating the concepts having the highest energy savings potential into the baseline system. Weight and energy savings gained by employing these concepts are listed in Table 68. Combined, they provide a fuel savings of 3.06 M-lb per aircraft over its life. This represents a savings of 28% over the baseline. A major portion of this gain is the result of weight reduction. The target system equipment weight is 384 lb less than the baseline -- a reduction of 14.5%. As discussed in Section 2.1, there is a relationship between weight and energy (fuel). Using fuel consumption coefficients, an equivalent weight of 868 lb was computed. This value includes equipment weight plus growth factor, and fuel weight savings due to the weight reduction and lower power usage. Eight hundred sixty-eight pounds represents 32.3% of the baseline hydraulic system weight. This significant weight reduction could be used to increase aircraft performance, either dynamically or in endurance, or in heavier weapon stores which would reduce the number of aircraft required.

TABLE 68. Target system weight savings

	WEIGHT REDUCTION (LB)	ENERGY REDUCTION (M-LB FUEL)
HCV Pump	6	0.450
Distribution System	35	0.022
Metal Bellows Accumulator	33	0.120
Nonlinear Valves	0	0.373
Dual Pressure Level System	-8	0.460
Control System	0	0.210
Advanced Materials	318	1.430
Total	384 LB (14.5%)	3.065 M-LB (28%)

EQUIVALENT WEIGHT REDUCTION 868 LB (32.3%)

2.7 STUDY PHASE CONCLUSIONS

The most promising energy savings techniques, based upon total energy consumption and using the Figure of Merit rating method, are listed below in descending order of potential.

CANDIDATE CONCEPTS	FOM RATING
Advanced Materials	1.30
Dual-Pressure System	0.41
Pumps	0.37
Non-Linear Valves	0.36
Distribution System	0.34
Variable Gain/Bandwidth	0.19
Accumulators	0.14
Hybrid Hyd/EM	0.07
Flow Augmentation	0/+0.50

The best approach, by far, is utilization of advanced materials to reduce weight and fuel consumption. Dual-pressure systems were next best. Application of all these concepts to the baseline vehicle produces appreciable savings: weight is reduced 14%, fuel consumption is decreased 28%. These savings are equivalent to 868 lb in weight.

The study demonstrates the necessity of considering the aircraft and hydraulic system on a global basis. Weight was shown to dominate the factors affecting total fuel consumption. Energy savings achieved by using a more efficient component can easily be negated if the component increases total weight -- even a few pounds. Control laws used to design the flight control system impact the power requirements of the hydraulic system. Control law development for new aircraft must, therefore, consider this impact on hydraulic system life and energy consumption; the design must be on a total systems basis.

After weight, hydraulic pumps are the next largest consumer of power. Pump losses are relatively constant and essentially independent of pump output. Losses are generally 12% to 18% of the peak output power. Peak power is determined by the peak load demand; reduction of peak load demand reduces pump size (weight) and losses, both of which decrease energy consumption. Any energy reduction concept which lowers the peak demand also effects savings in the hydraulic distribution system -- direct and indirect savings.

Energy savings translate into cost savings. Assuming a fuel cost of 10¢ per pound, the target system would use $236,000 less fuel per aircraft life than the baseline vehicle. Assuming a fleet of 500 aircraft, the dollar savings would be $118,000,000. This represents a 3.3% cost savings over a fleet of baseline aircraft. The "real" savings, however, lie in the equivalent weight savings of 868 lb/aircraft. This translates into a heavier stores capability, increased performance, and extended missions. Considering stores, the baseline load was 6,400 lb; the target system aircraft could carry 7369 lb, an increase of 15%. For an air-to-ground attack mission, 15% fewer aircraft could deliver the same amount of stores. This represents savings in human life, equipment life and operational costs. The effects of weight reduction are, therefore, even more important than fuel cost savings.

Observations reached in this study, are summarized in Figure 72. The final conclusion was clear:

"Don't add weight to save energy"

OBSERVATIONS

- Weight Dominates
- Pumps Next Largest Consumer
- Electrical Is Minor
- System Components Are Sized by Peak Loads. Reduction in Peaks Reduces System Weight and Save Energy
- Target System Saves $236K Over Life of Aircraft (@ 10¢/Lb Fuel Cost)
- Real Savings in Energy Lies in the Equivalent Weight It Saves. (i.e. Increases Performance, Extended Mission, Etc)

CONCLUSION

Don't Add Weight to Save Energy

Figure 72. Observations and conclusion

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ABBREVIATIONS AND SYMBOLS

Abbreviations or Symbol	DEFINITION
A	piston net area
A/C	aircraft
ACC	accumulator
ACT.	actuator
AFCS	automatic flight control system
AMAD, AM	aircraft mounted accessory drive
APU	auxiliary power unit
ASUW	air-to-surface warfare
ATA	Advanced Tactical Aircraft (Navy)
ATF	Advanced Tactical Fighter (Air Force)
B	backup
CBW	control-by-wire
cc/m	cubic centimeters per minute
C_D	drag coefficient
C_L	lift coefficient
db	decibel
DDV	direct drive valve
DEG	degree
DL	design load
D_m	actuator displacement, inches
DSN	design
ECS	environmental control system
EDU	electronic drive unit
EHA	electro-hydrostatic actuator
EHV	electro-hydraulic servo valve
EM	electro-mechanical

Abbreviations or Symbol	DEFINITION
ES	energy savings
FBW	fly-by-wire
F/C	flight control
FCR_{HP}	fuel consumption rate per horsepower coefficient
FCR_{LB}	fuel consumption rate per pound of equipment weight coefficient
FOM	figure of merit
ft	feet
g	gust
gal	gallon
HCV	hybrid check valve (pump)
H_M	hinge moment
hp	horsepower
hr	hour
HVDC	high voltage direct current
Hz	Hertz (cycles per second)
IAP	integrated actuator package
I.D.	inside diameter
in^3	cubic inches
INBD	inboard
IRT	intermediate rated thrust
K	thousand
K_p	valve pressure gain coefficient
K_q	valve flow gain coefficient
KW	kilowatts
lb	pound
L/D	lift-to-drag ratio
LE	leading edge
LH	left hand (side)
LHS	lightweight hydraulic system

Abbreviations or Symbol	DEFINITION
M, M_N	mach number
MIN	minimum or minute (time)
M-lb	10^6 x pounds
MTBF	mean-time-between failures
MX	maximum
N	normal
NL	no-load
NM	nautical mile
NO.	number (quantity)
NOM	nominal
OAB	outer air battle
O.D.	outside diameter
OTBD	outboard
P	pressure
PI	pressure intensifier
PM	powder metallurgy
psi	pounds per square inch
PWGF	power growth factor
Q	flow
RCS	radar cross-section
rev	revolution
RH	right hand (side)
RN	Reynold's number
RVDT	rotary variable differential transformer
SAS	stability augmentation system
sec	second (time)
SFC	specific fuel consumption
SHV	shuttle valve
S.L.	sea level

Abbreviations or Symbol	DEFINITION
SME	subject matter experts
STAB	horizontal stabilizer
t	time
T	torque or time
t/c	wing thickness-to-cord ratio
TE	trailing edge
T/R	thrust reverser
T/V	thrust vectoring
VFMX	Advanced Multimission Fighter/Attack (Navy)
VHP	very high pressure
VOL	volume
W	work
WT	weight
WTGF	weight growth factor
Δ	delta
η	efficiency
$\dot{\theta}$	control surface angular rate
μ	micro (10^{-6})
ω	frequency (radians per second)

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APPENDIX A

INDUSTRY SURVEY

Contents

SURVEY QUESTIONNAIRE

Yes No 1. Could heat rejection in your pumps be reduced if special effort were applied in this area? If so, please outline possible approaches and estimate their potential energy savings.

Yes No 2. Are you developing computer-controlled pumps to provide the capability of power matching? If so, please outline your design approaches.

Yes No 3. Are you developing energy efficient components suitable for use in integrated actuator packages, such as servo pumps and variable displacement hydraulic motors, etc.? If so, please explain basic design features of your hardware.

Yes No 4. Are you pursuing development of other components or system concepts which either reduce power consumption, reduce weight, or improve performance of aircraft hydraulic systems? If so, please provide details.

SURVEY QUESTIONNAIRE

Yes No 1. Are you developing direct drive servo valves or other actuation controls which have the potential of reducing energy consumption, reducing weight or improving performance? If so, please provide details.

Yes No 2. Are you applying developing computer controlled servo valves to reduce leakage, improve performance or manufactorability? If so, please describe.

Yes No 3. Are you applying advanced materials to actuators or other components to reduce weight or improve performance? If so, please describe approach.

Yes No 4. Are you developing rotary hingeline actuation compatible with flight control requirements of advanced aircraft? If so, please provide details.

Yes No 5. Are you pursuing development of other components or system concepts which either reduce power consumption, reduce weight, or improve performance of aircraft hydraulic systems? If so, please provide details.

COMPANIES SURVEYED

COMPANY	TELEPHONE NUMBER
Abex Corporation Aerospace Division 151 West 5th Street Oxnard, CA 93030	(805) 985-0217
Allied Bendix Aerospace Bendix Fluid Power Division 211 Seward Avenue Utica, NY 13503	(315) 797-2500
Allied Bendix Aerospace Bendix Electrodynamics Division 11600 Sherman Way North Hollywood, CA 91605	(213) 877-2881 (818) 765-1010
Crane Company Hydro-Air Division 3000 Winona Avenue Burbank, CA 91510	(213) 849-1331 (213) 842-6121
E-Systems, Inc. Montek Division 2268 South 3270 West Salt Lake City, UT 84119	(801) 484-8661

COMPANIES	TELEPHONE NUMBER
Garrett Corporation Aero Hydraulics, Inc. 5841 North West 9th Avenue Fort Lauderdale, FL 33309	(305) 772-8080
General Signal Corporation New York Airbrake Starbuck Avenue Watertown, NY 13601	(315) 782-7000
HR Textron, Inc. 25200 West Rye Canyon Road Valencia, CA 91355	(805) 259-4030
Hydraulic Units, Inc. 1700 Business Center Drive Duarte, CA 91010	(818) 359-9211
Vickers, Inc. 5353 Highland Drive Jackson, MS 39206	(601) 981-2811
National Waterlift Division Pneumo Corporation 2220 Palmer Avenue Kalamazoo, MI 49001	(616) 384-3400

COMPANIES	TELEPHONE NUMBER
Parker Hannifin Corporation Parker Bertea Aerospace Group Control Systems Division 14300 Alton Parkway Irvine, CA 92718	(714) 833-3000
Parker Hannifin Corporation Aerospace Hydraulic Division 18321 Jamboree Blvd Irvine, CA 92715	(714) 851-3302
Parker Hannifin Corporation Fluid Power Pump Division Tsego, MI 49078	(616) 694-9411
Rexroth Mobile Hydraulics Division Wooster, OH 44691	(216) 263-3300
Sterer Engineering & Manufacturing Co. 4690 Colorado Blvd Los Angeles, CA 90039	(213) 245-7161
Sundstrand Corporation Sundstrand Aviation 747 Harrison Avenue Rockford, IL 61125	(815) 226-6000
Teledyne - Sprague Engineering 9300 South Vermont Avenue Gardena, CA 90247	(213) 327-1610

APPENDIX B

TRADE STUDY DATA

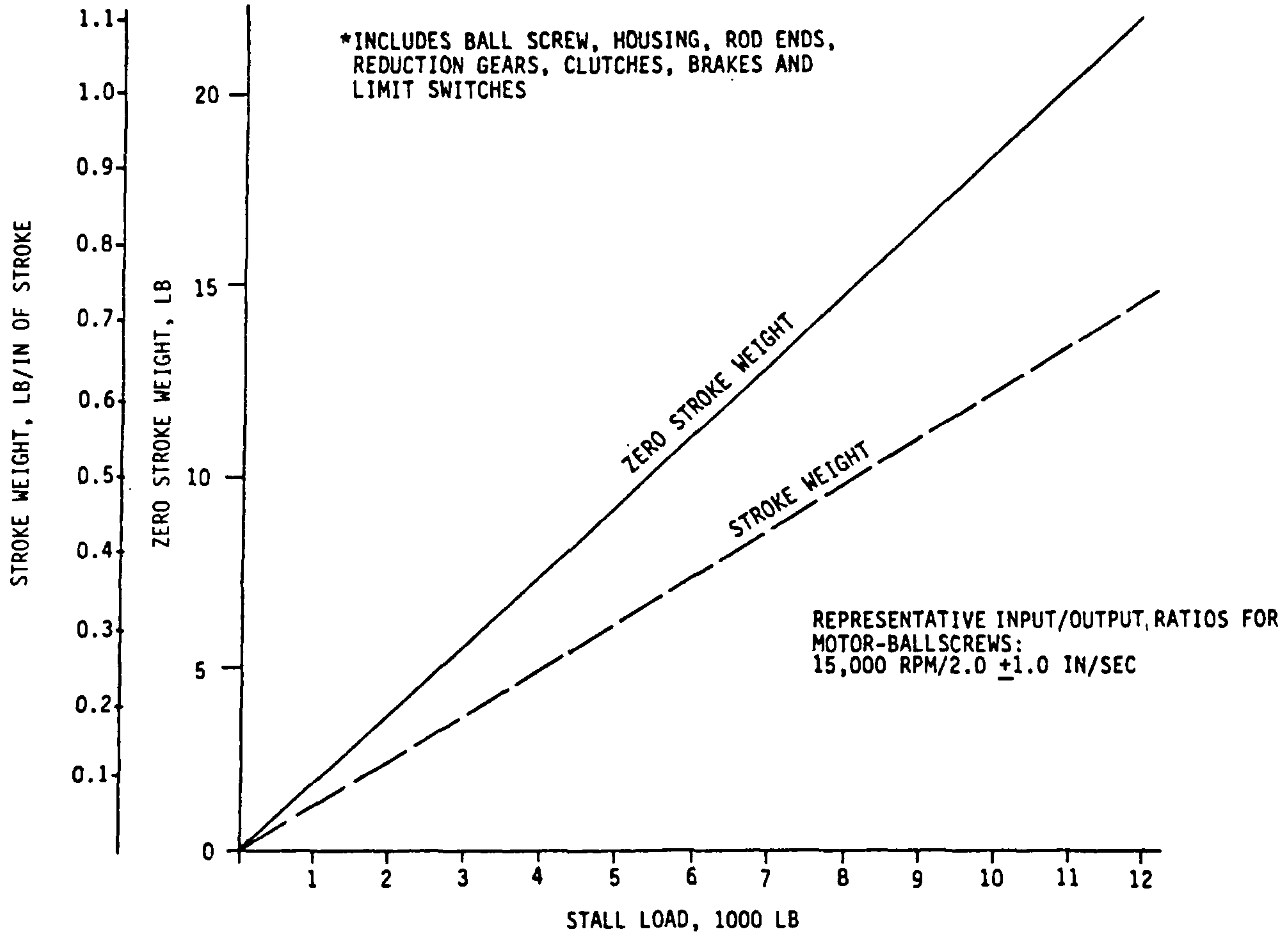


Figure B-1. Ballscrew actuator *assembly weight

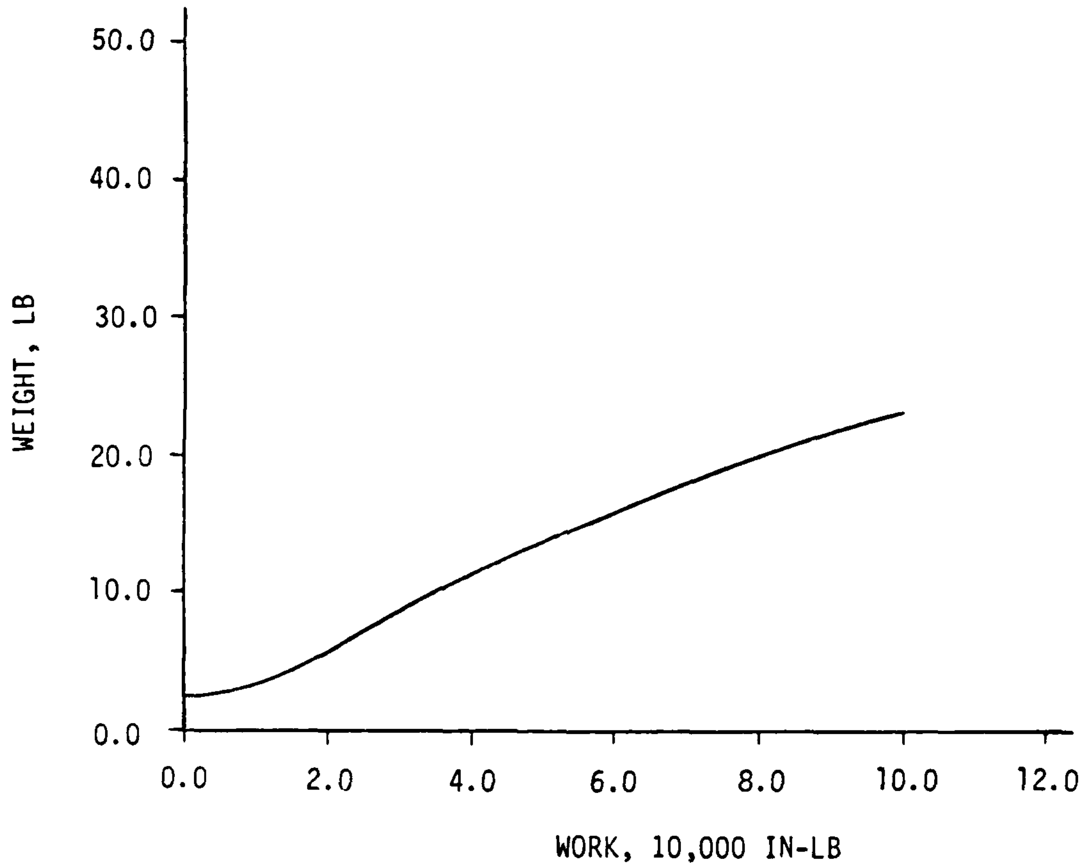


Figure B-2. Ballscrew weight vs. work

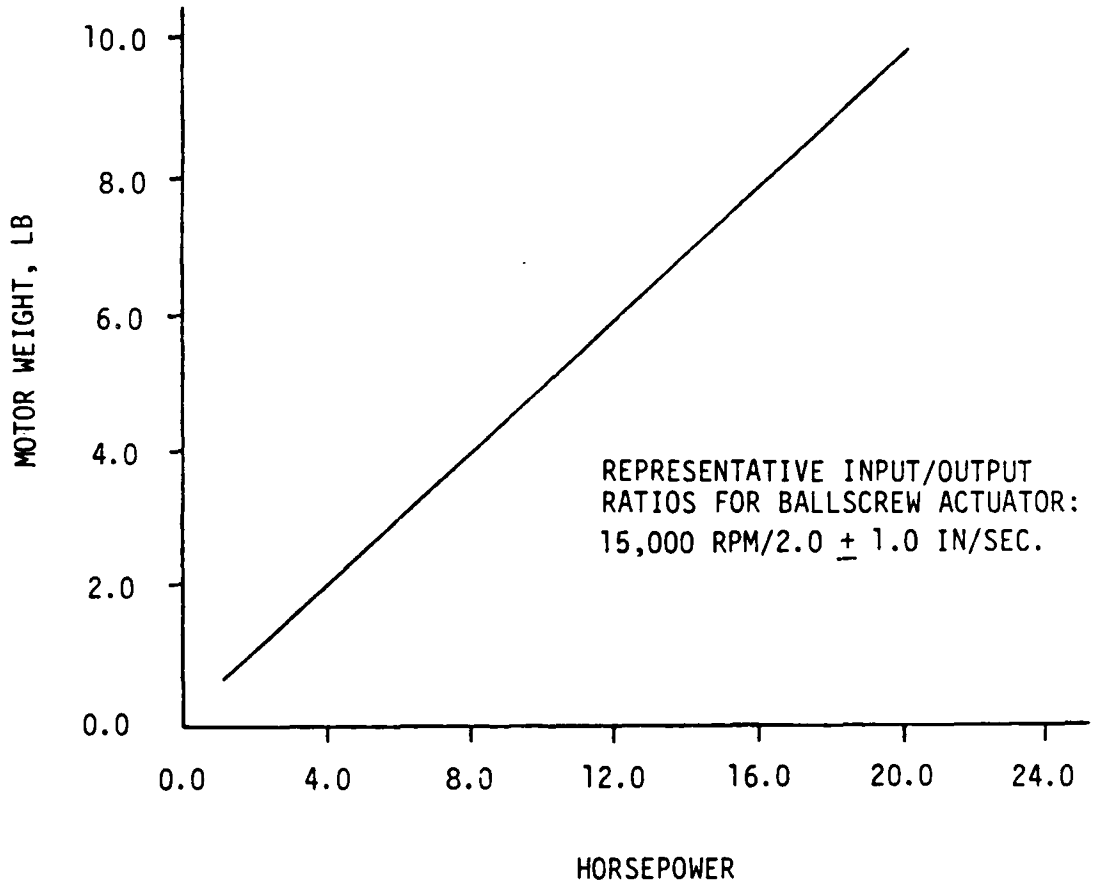


Figure B-3. Ballscrew motor weight vs. power

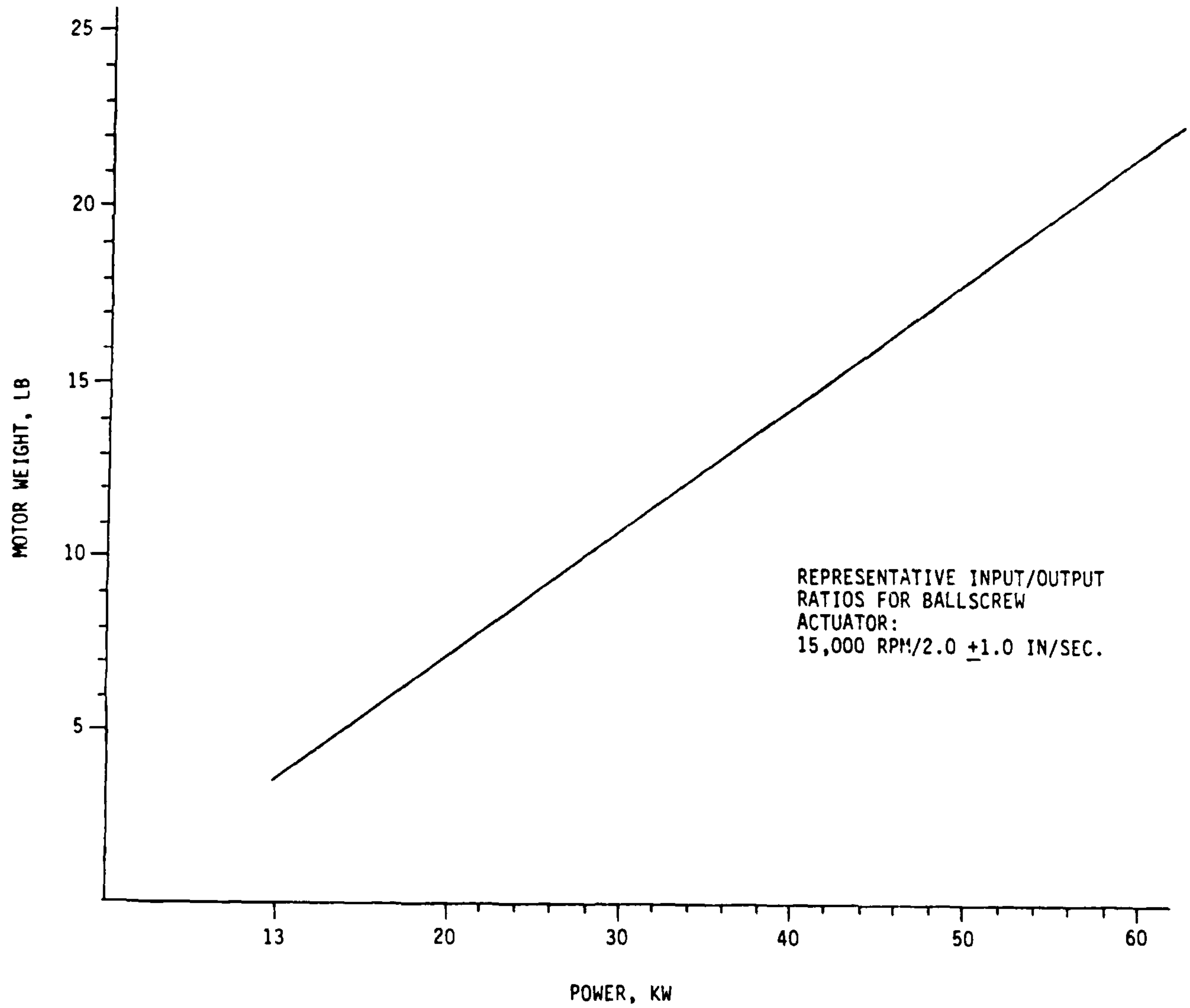


Figure B-4. Electric motor weight vs. power

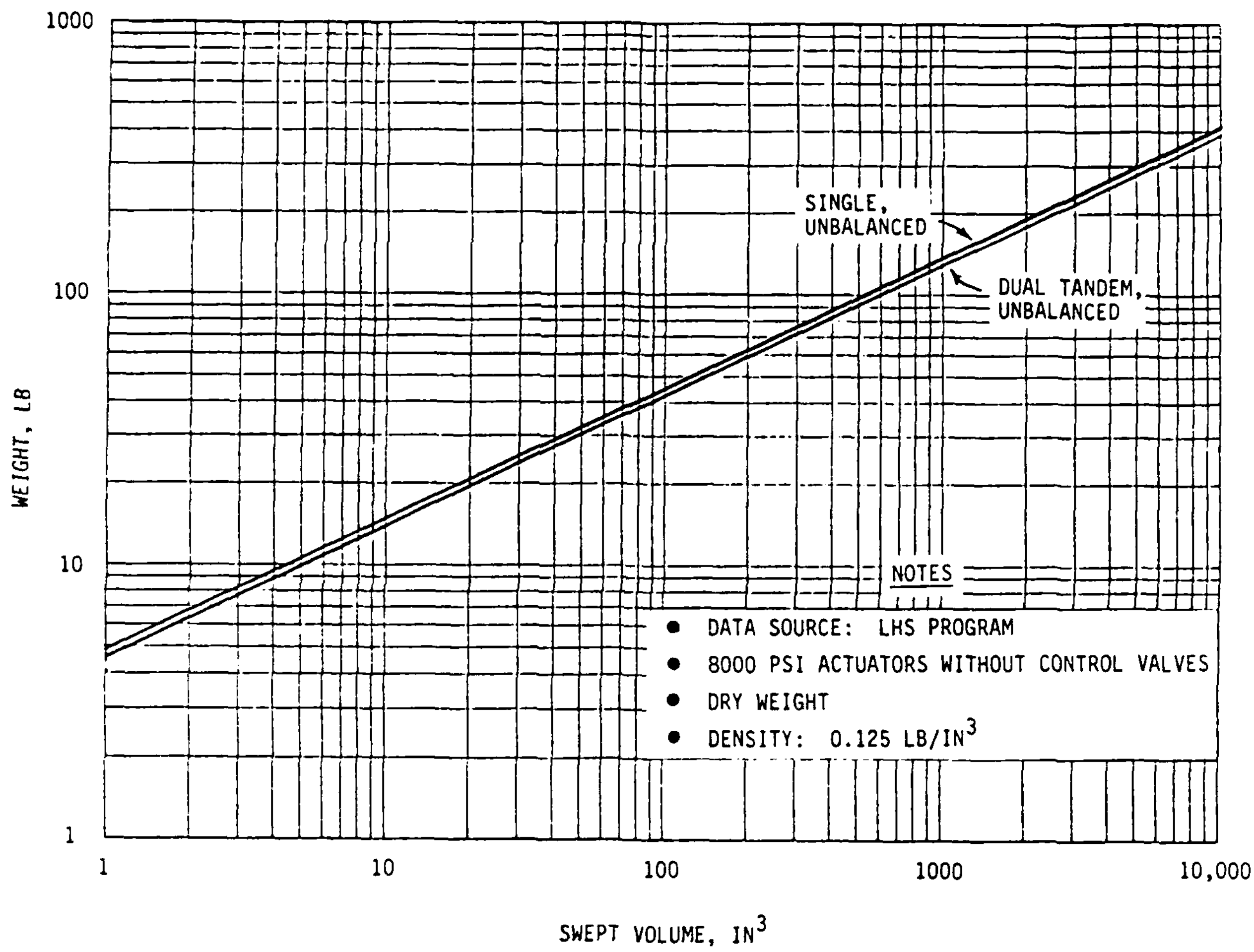


Figure B-5. Linear actuator weight

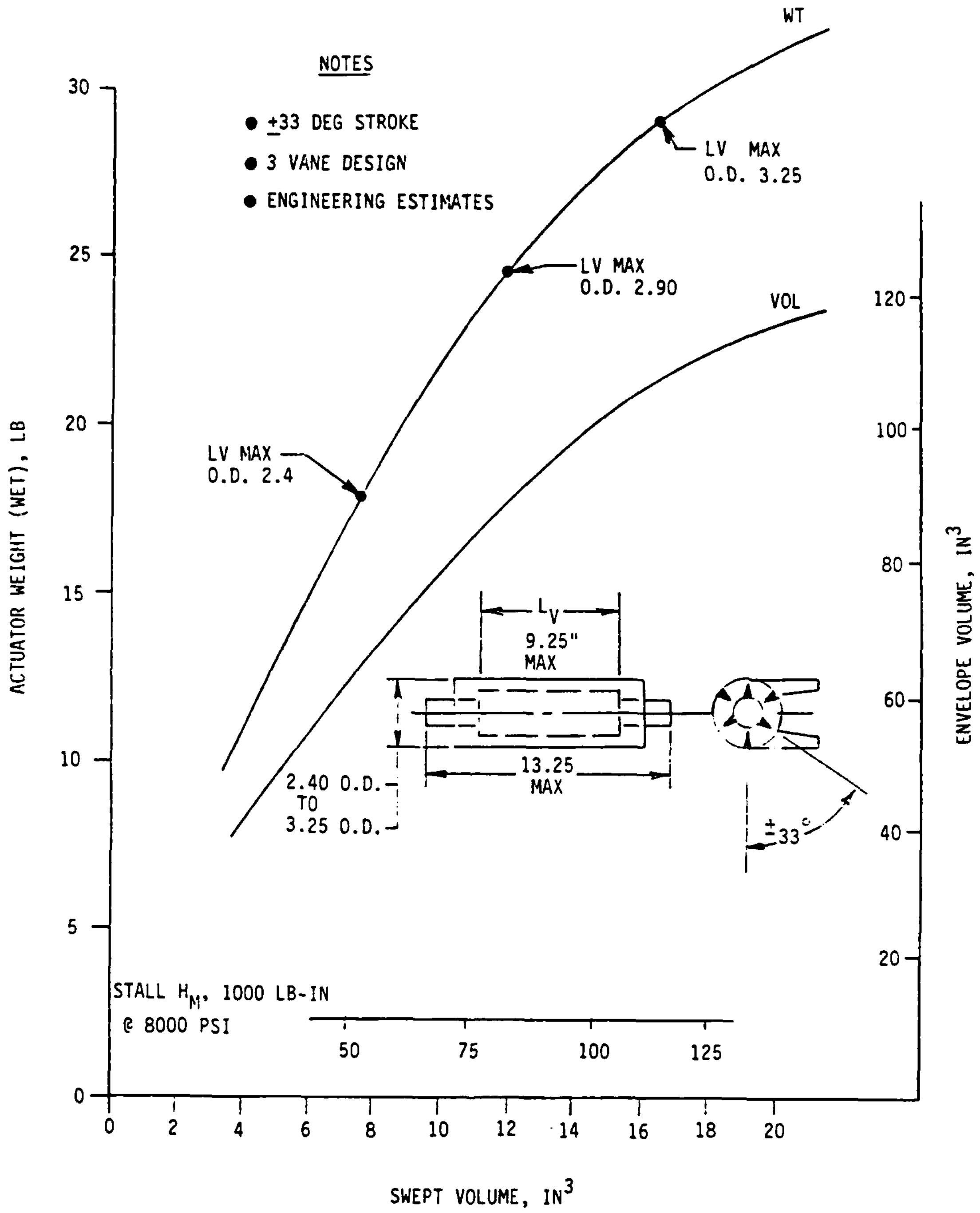


Figure B-6. Rotary actuator weight, 3 vane

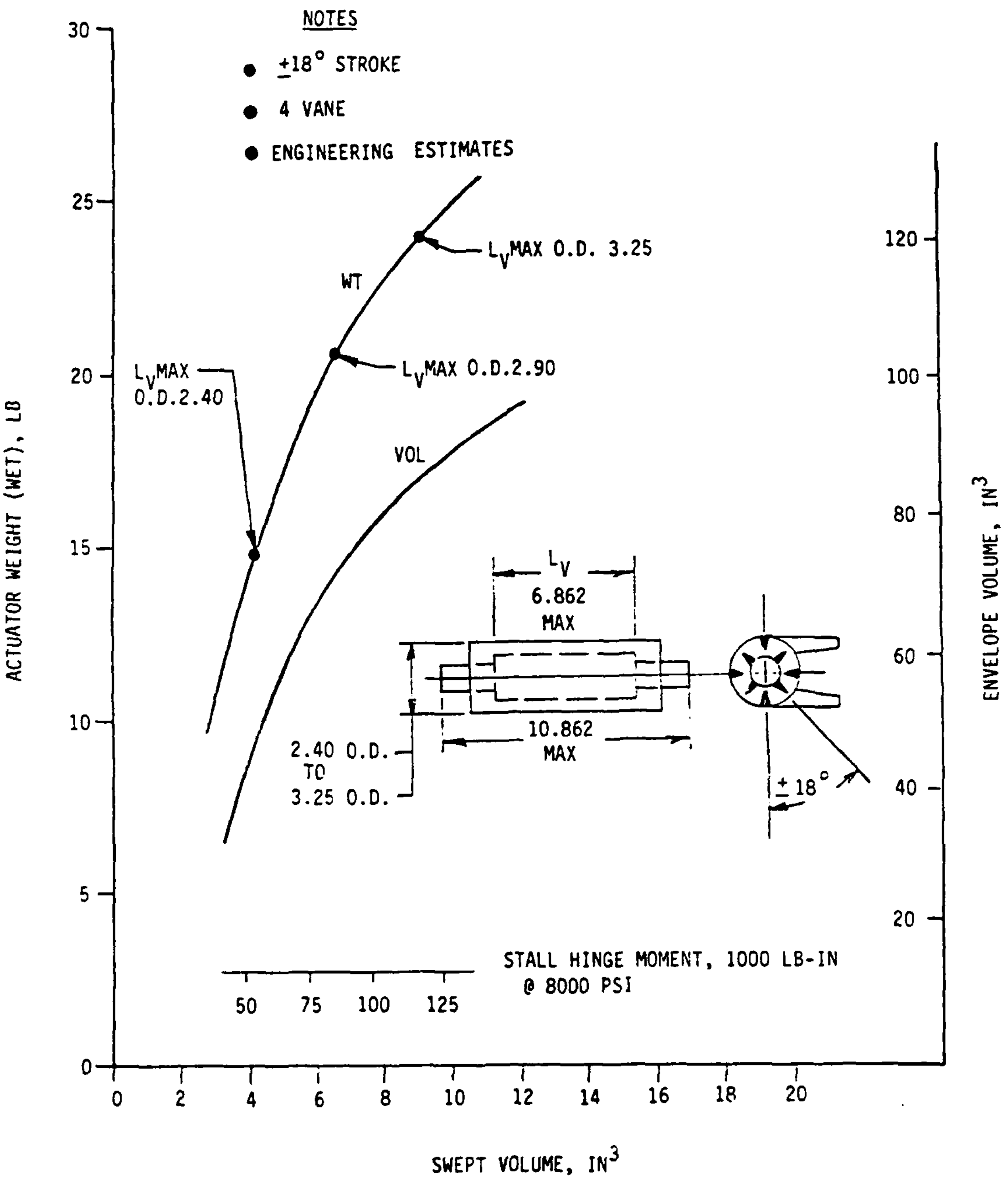


Figure B-7. Rotary actuator weight, 4 vane

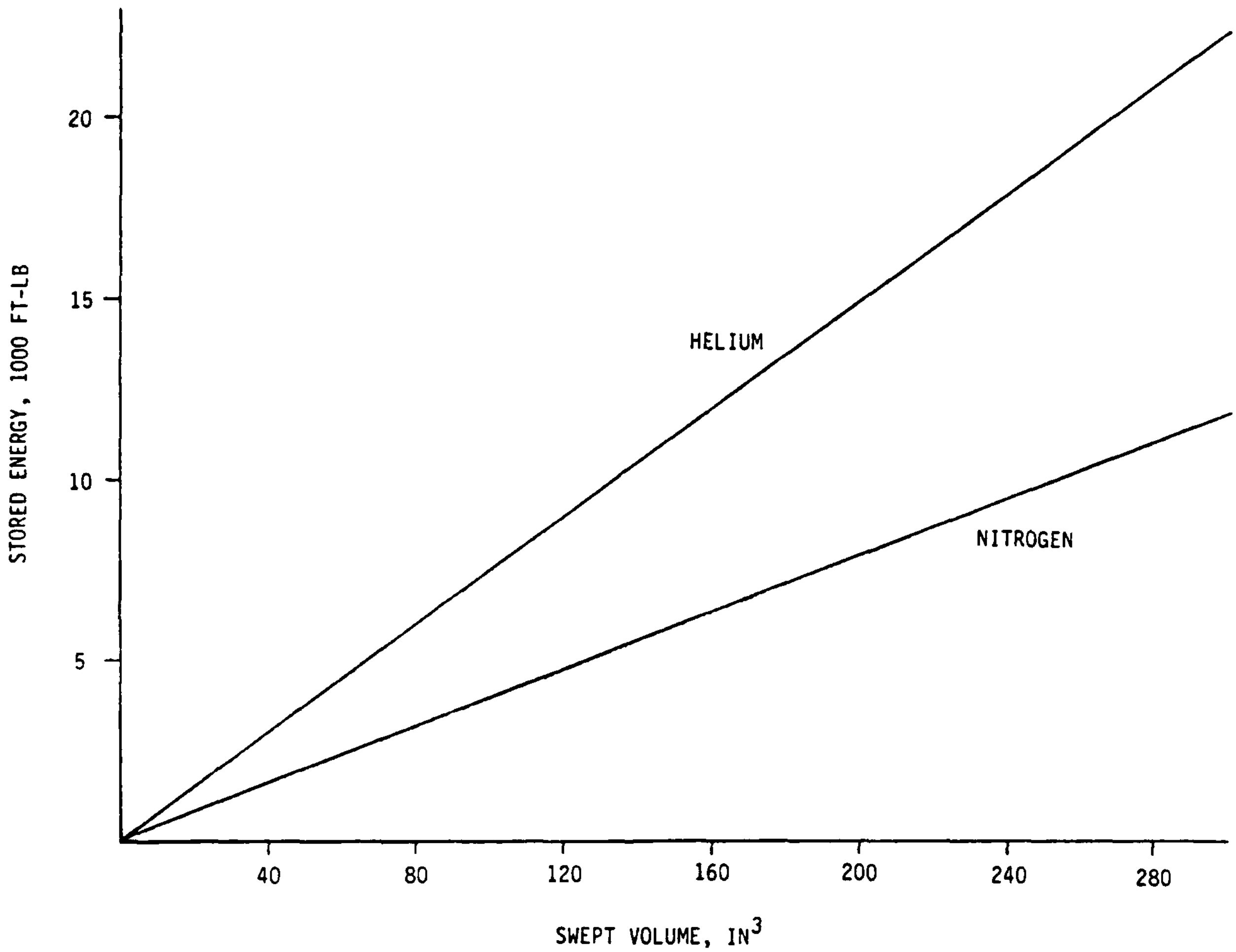


Figure B-8. Metal bellows accumulator, stored energy vs. volume

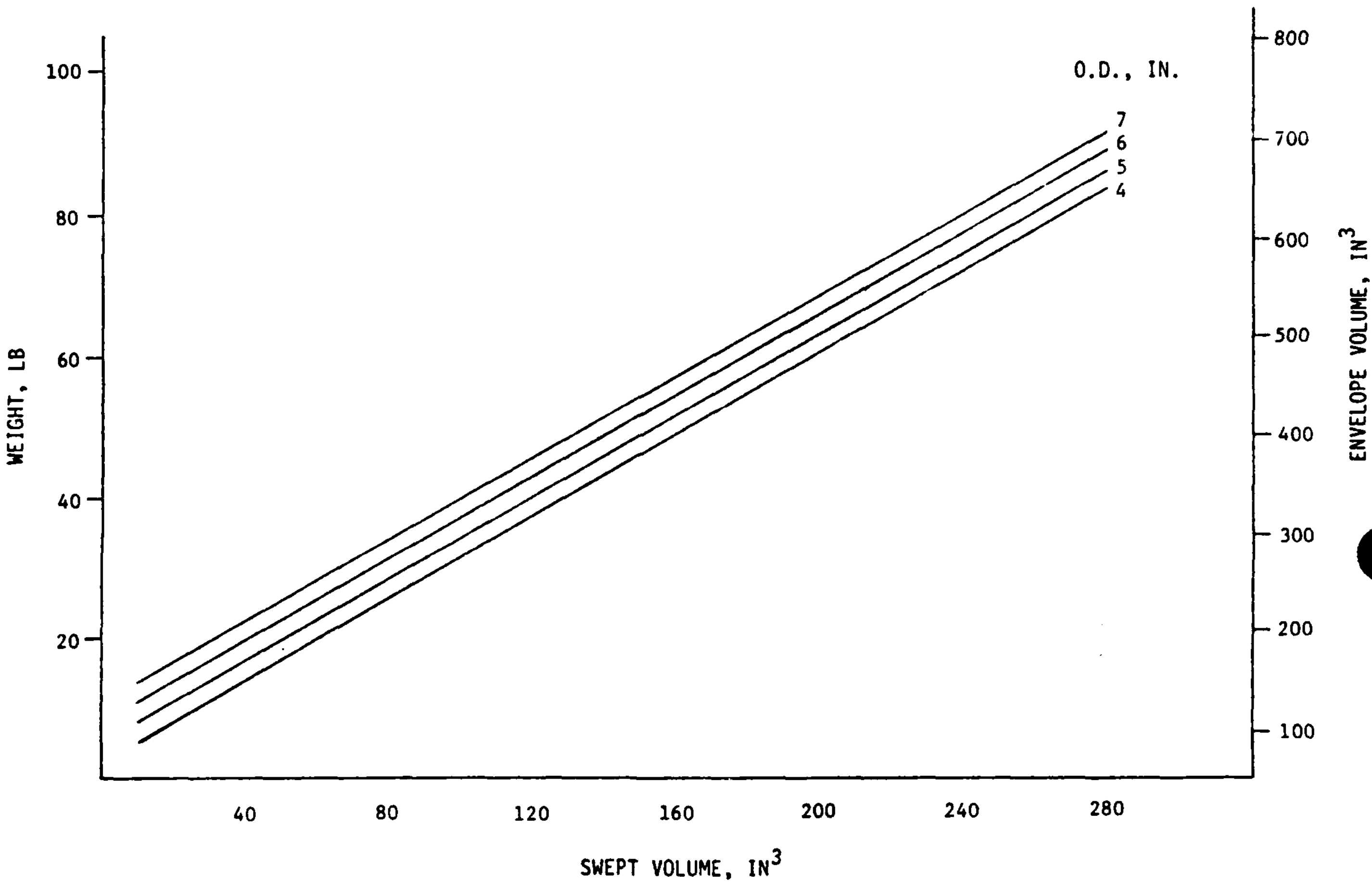


Figure B-9. Metal bellows accumulator, weight and volume vs. swept volume

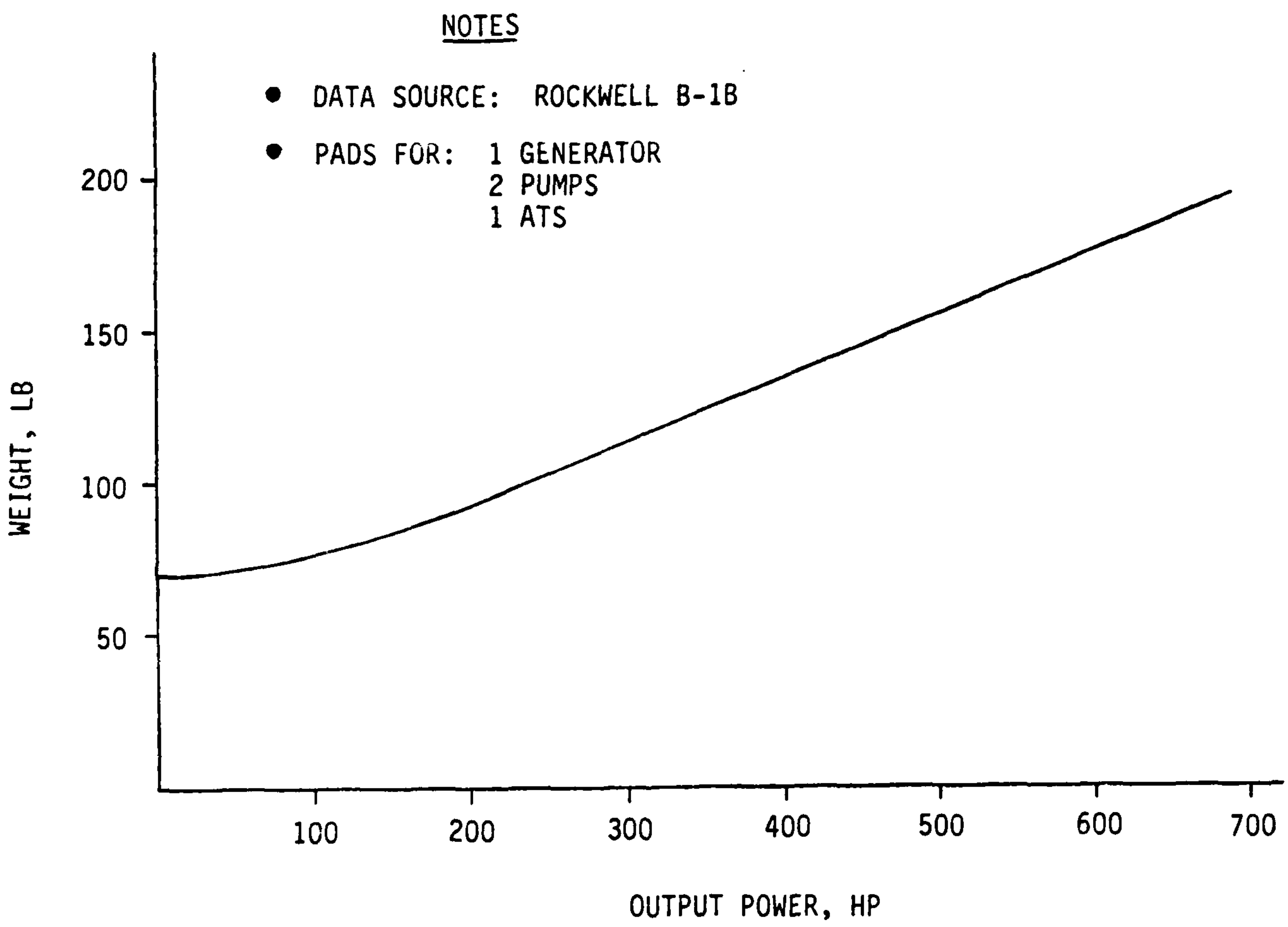


Figure B-10. AMAD weight vs. power

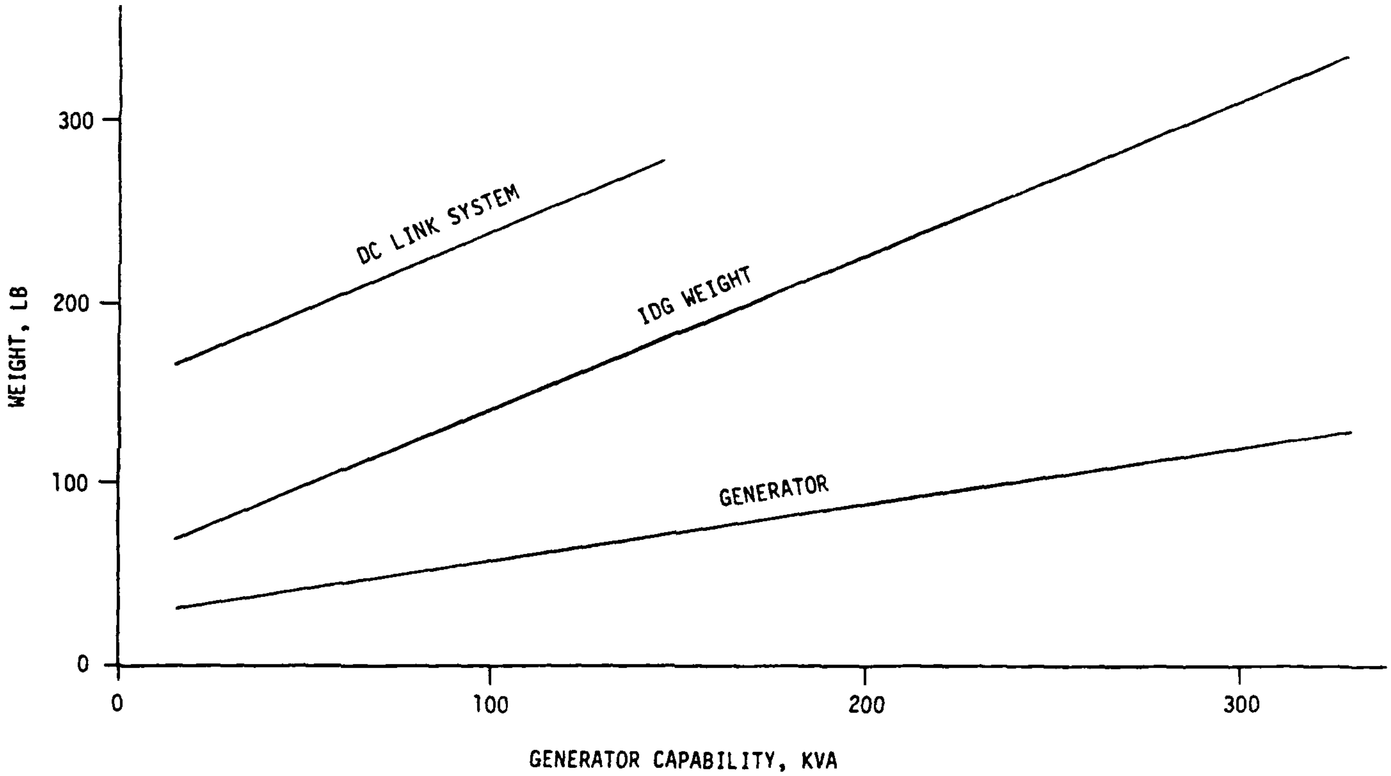


Figure B-11. Electrical power system weight vs. KVA

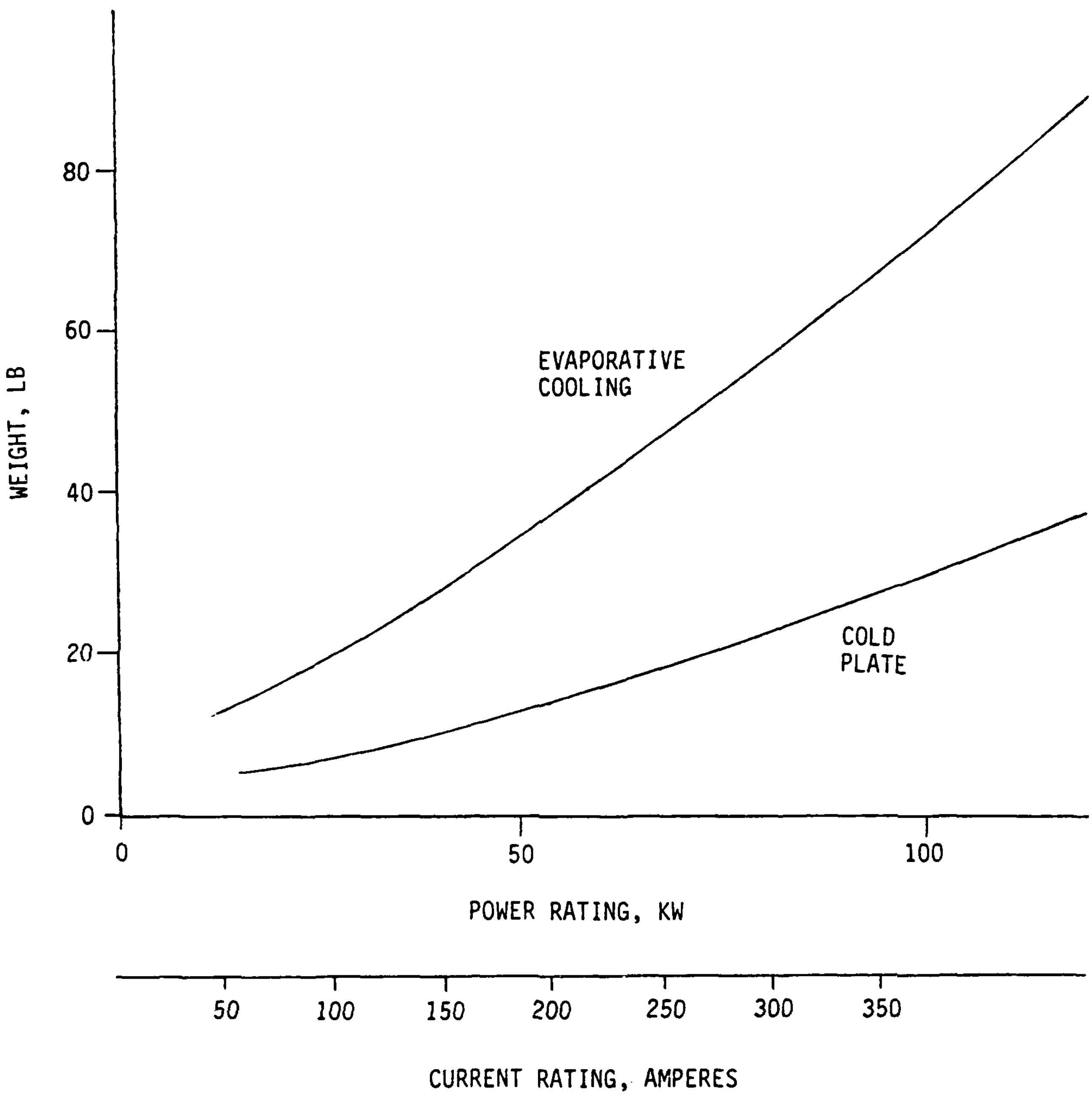


Figure B-12. Inverter weight vs. power

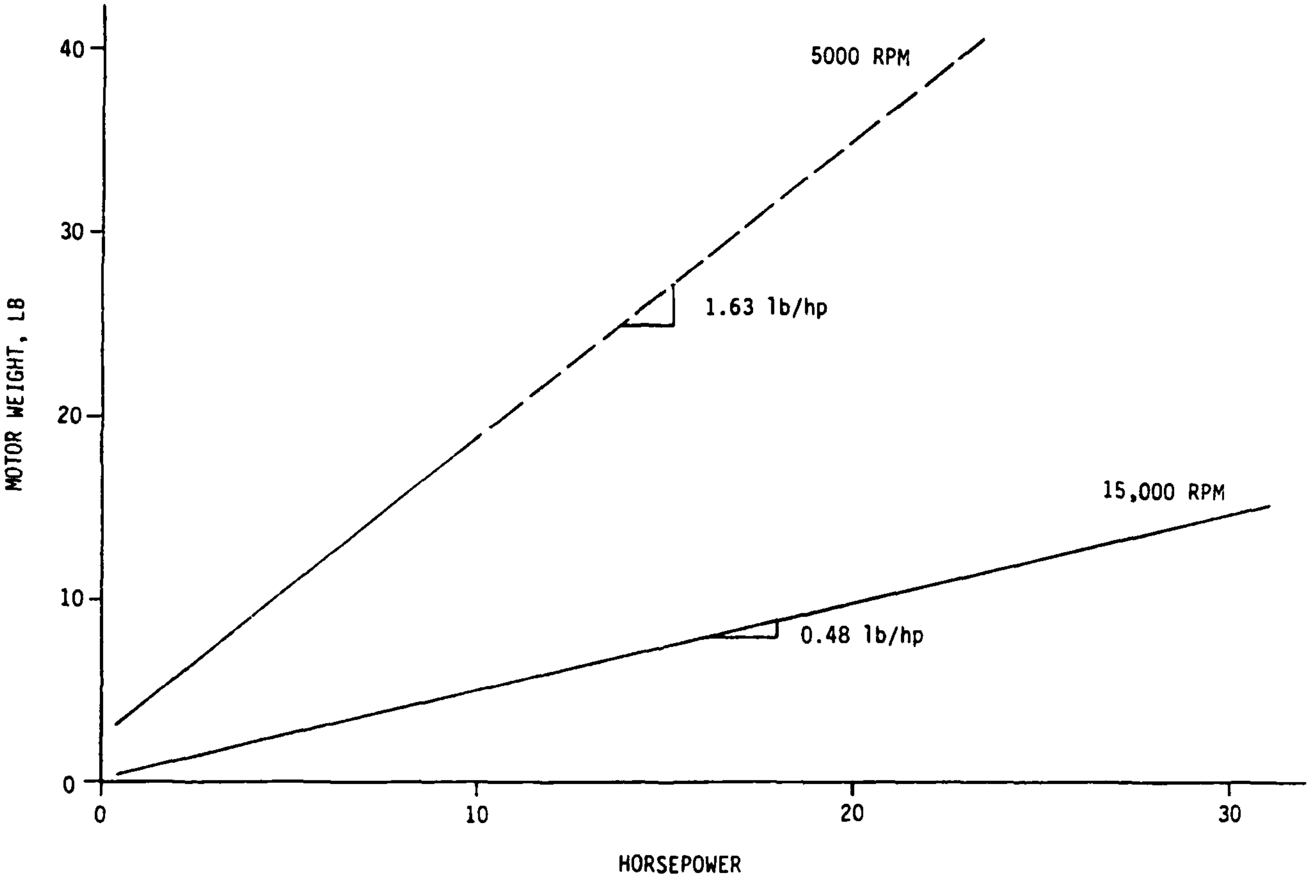
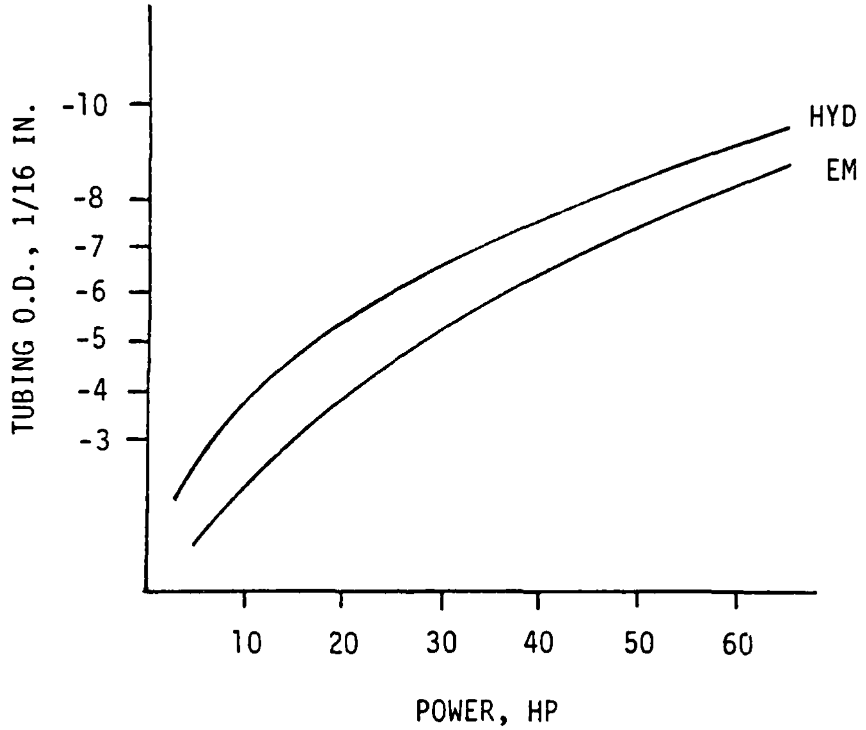


Figure B-13. 270 volt brushless DC motor weight vs. power

NOTES

HYDRAULIC: 8000 PSI SYSTEM
FLUID VELOCITY < 25 FT/SEC

ELECTRICAL: 270 VDC SYSTEM
INSULATION TEMPERATURE



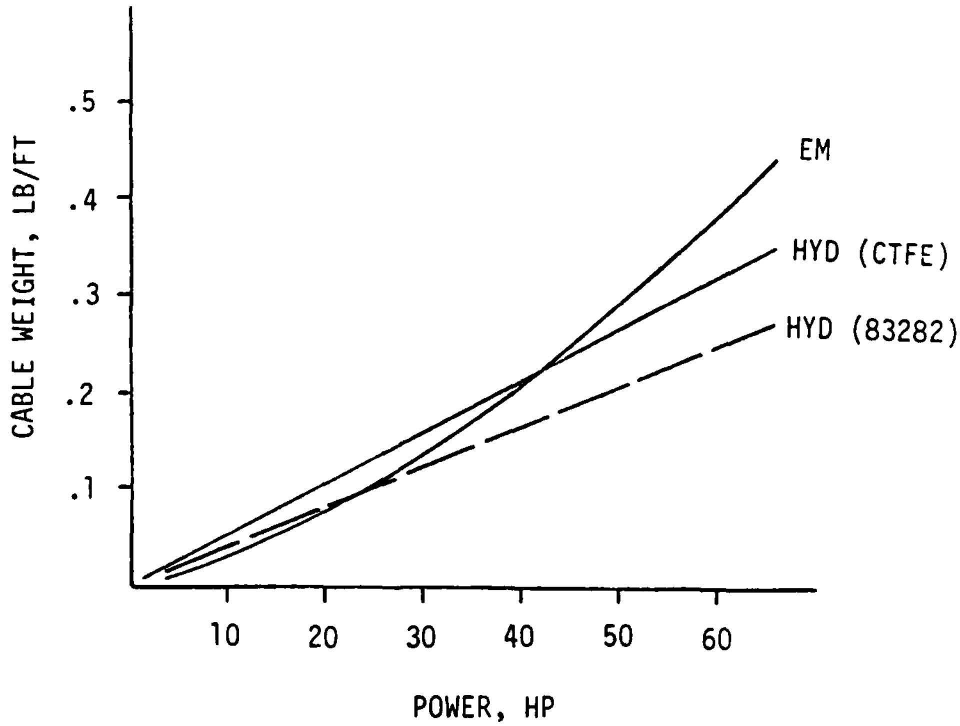


Figure B-14. Hydraulic vs. electrical power transmission

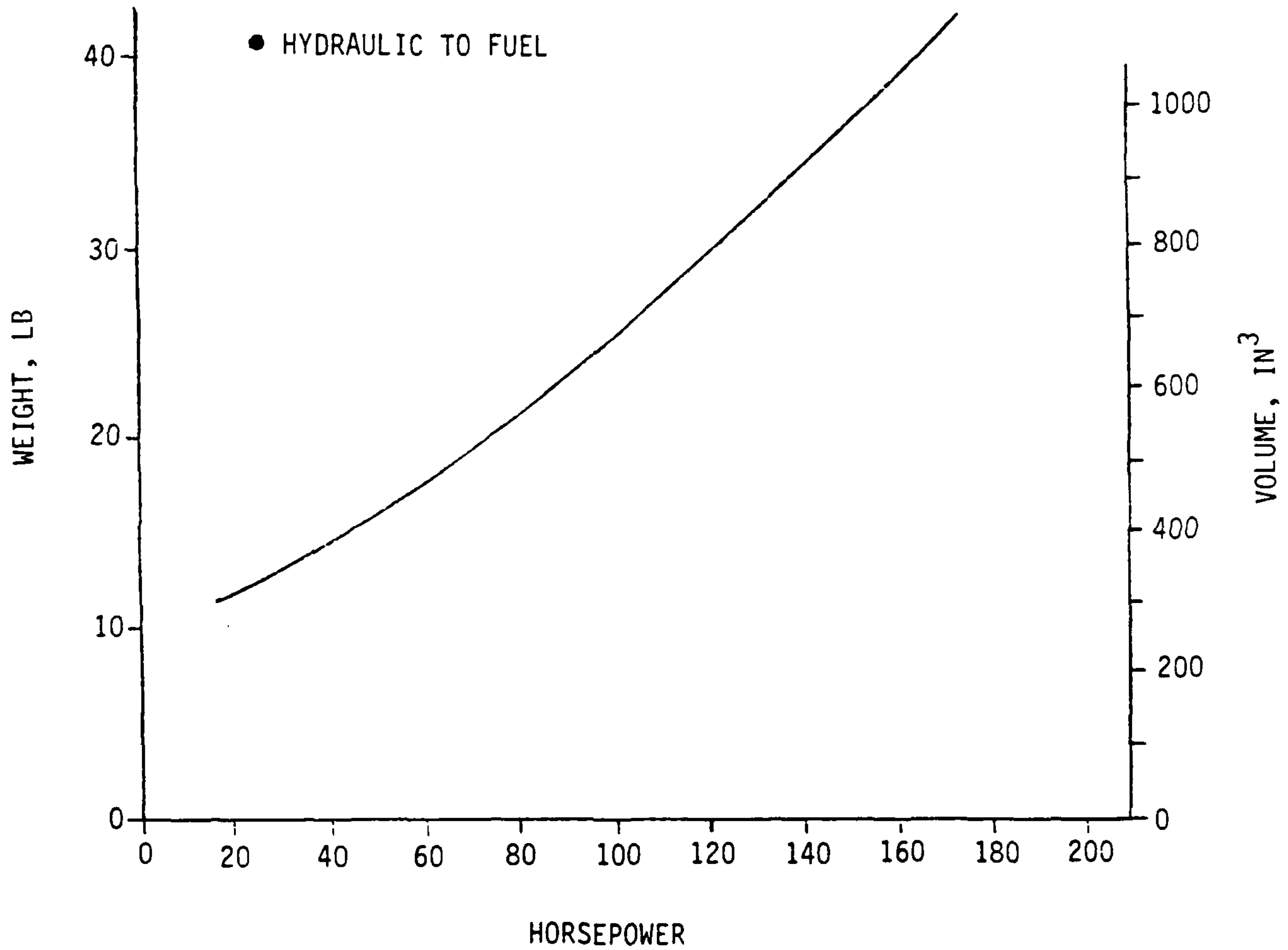


Figure B-15. Heat exchanger weight vs. power

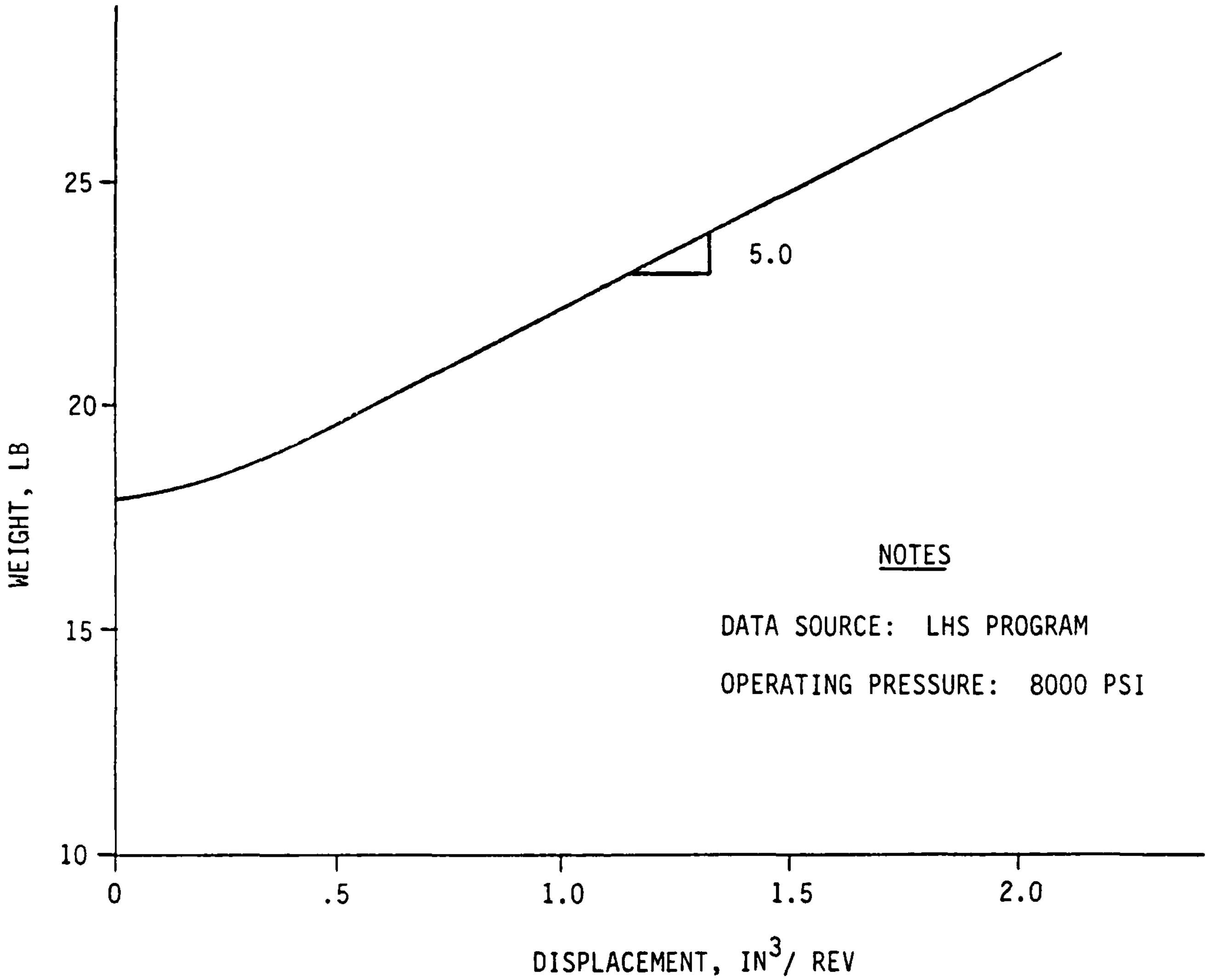


Figure B-16. Hydraulic pump/motor weight vs. displacement

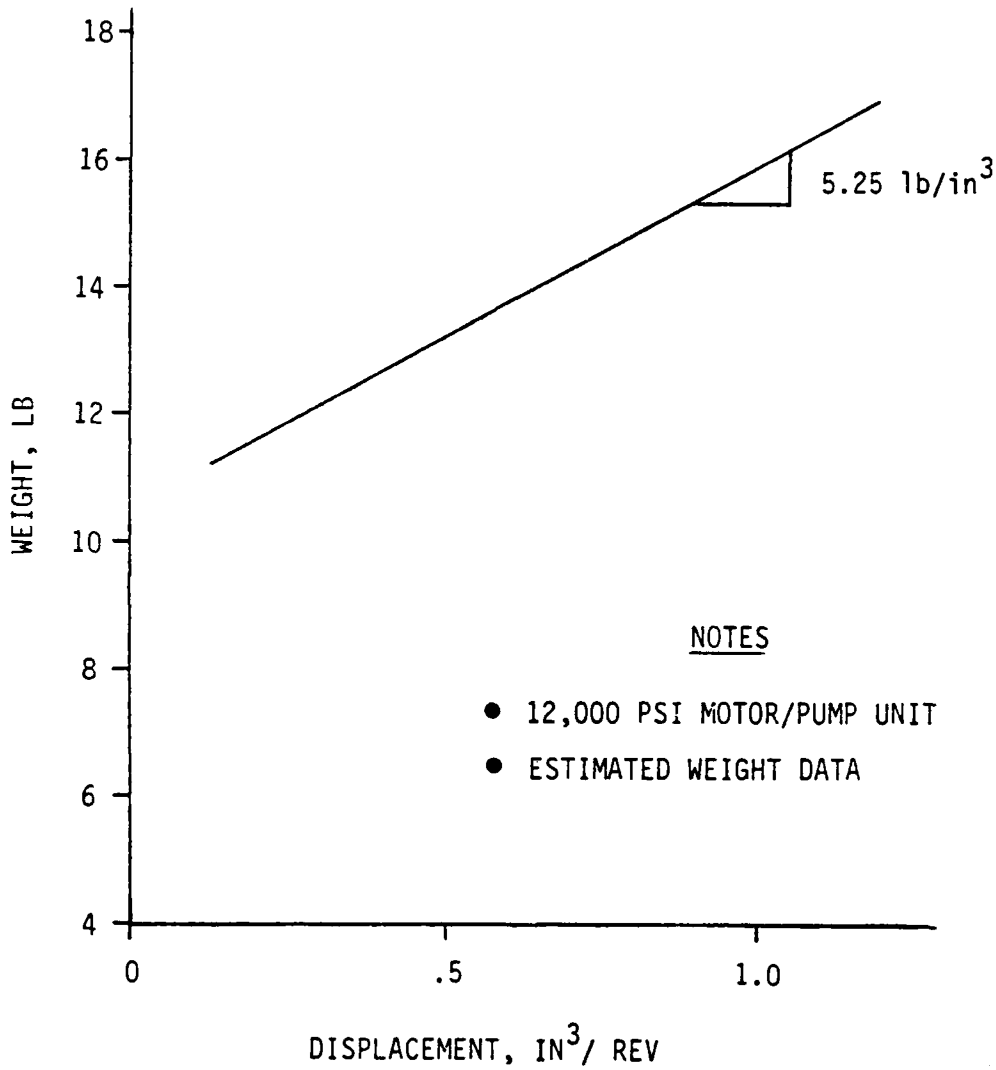


Figure B-17. Pressure intensifier weight vs. displacement

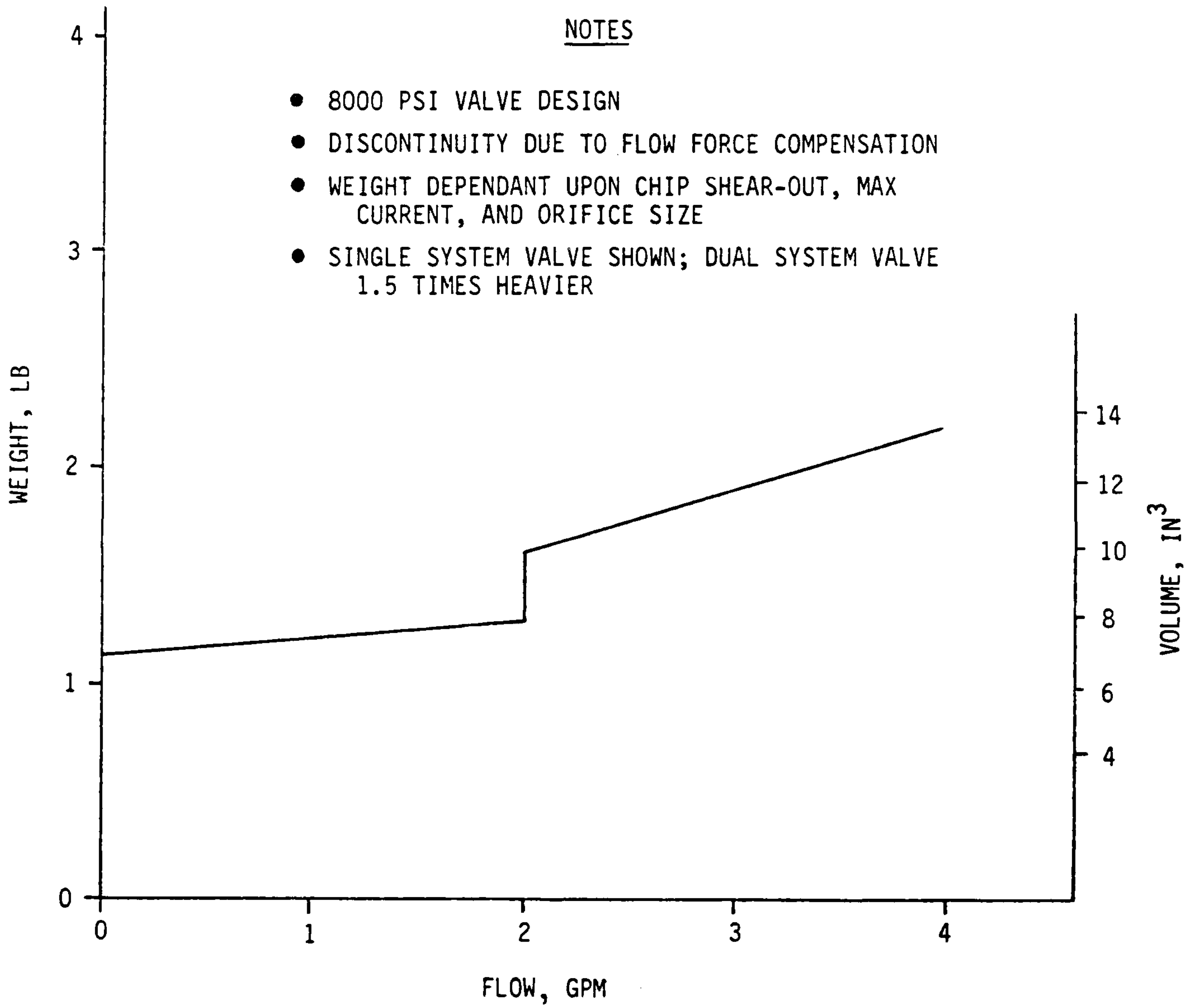


Figure B-18. Direct drive valve weight and volume vs. flow

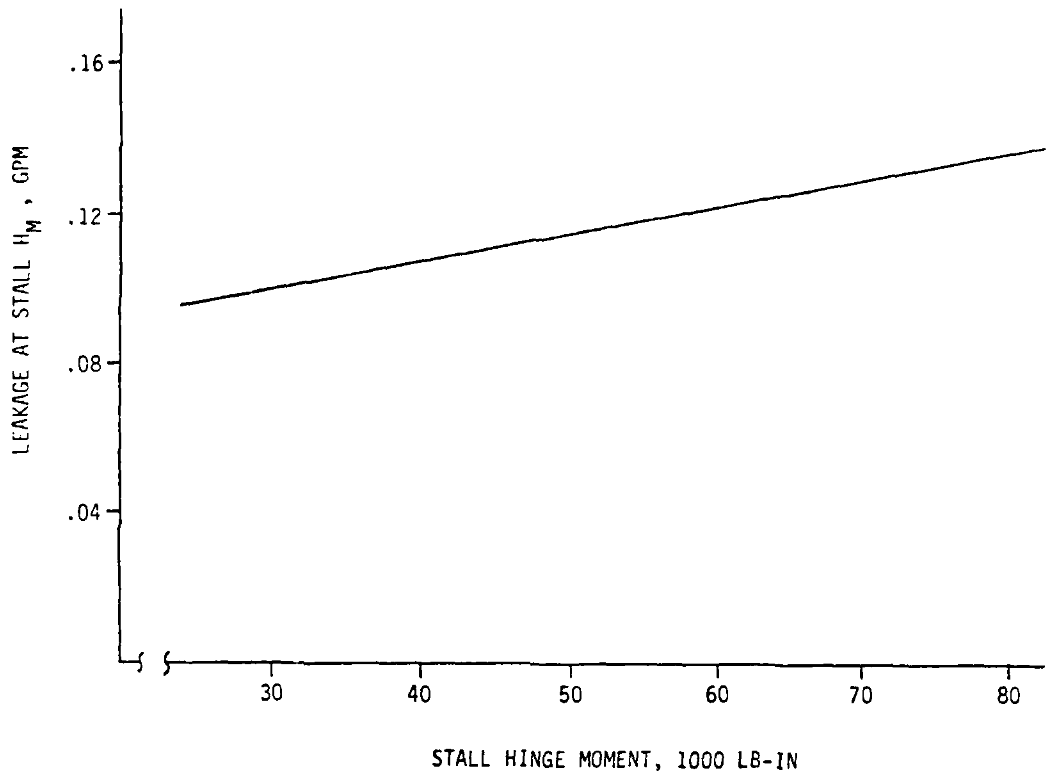


Figure B-19. Vane actuator leakage vs. hingemoment

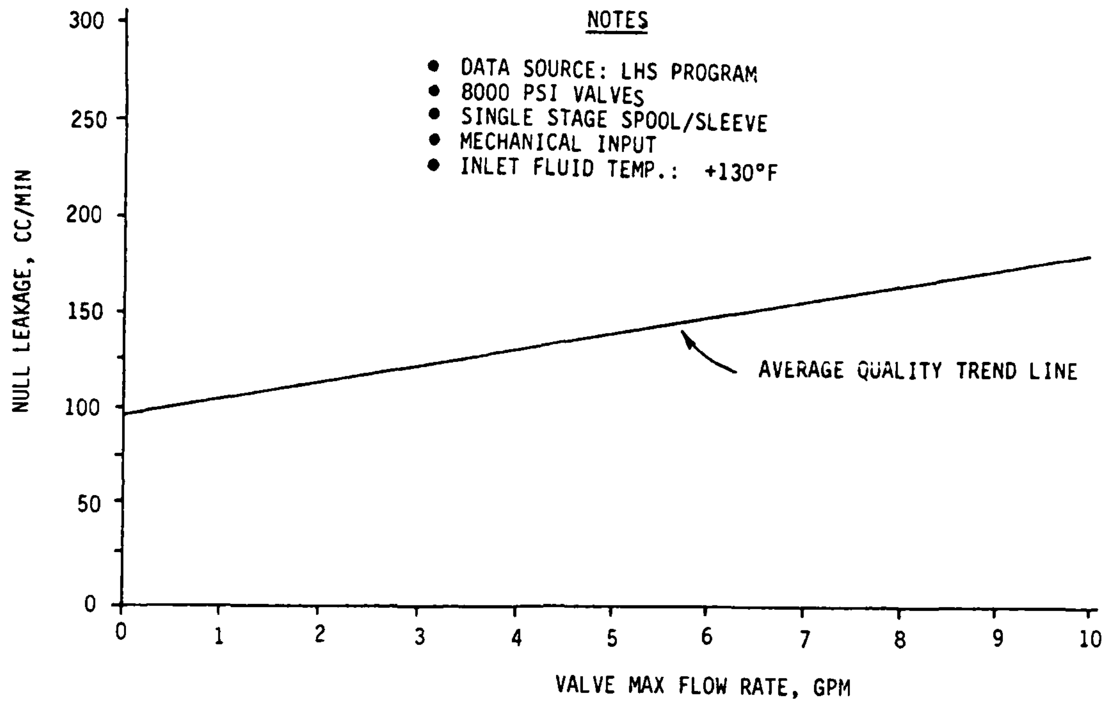


Figure B-20. Valve null leakage vs. no-load rate

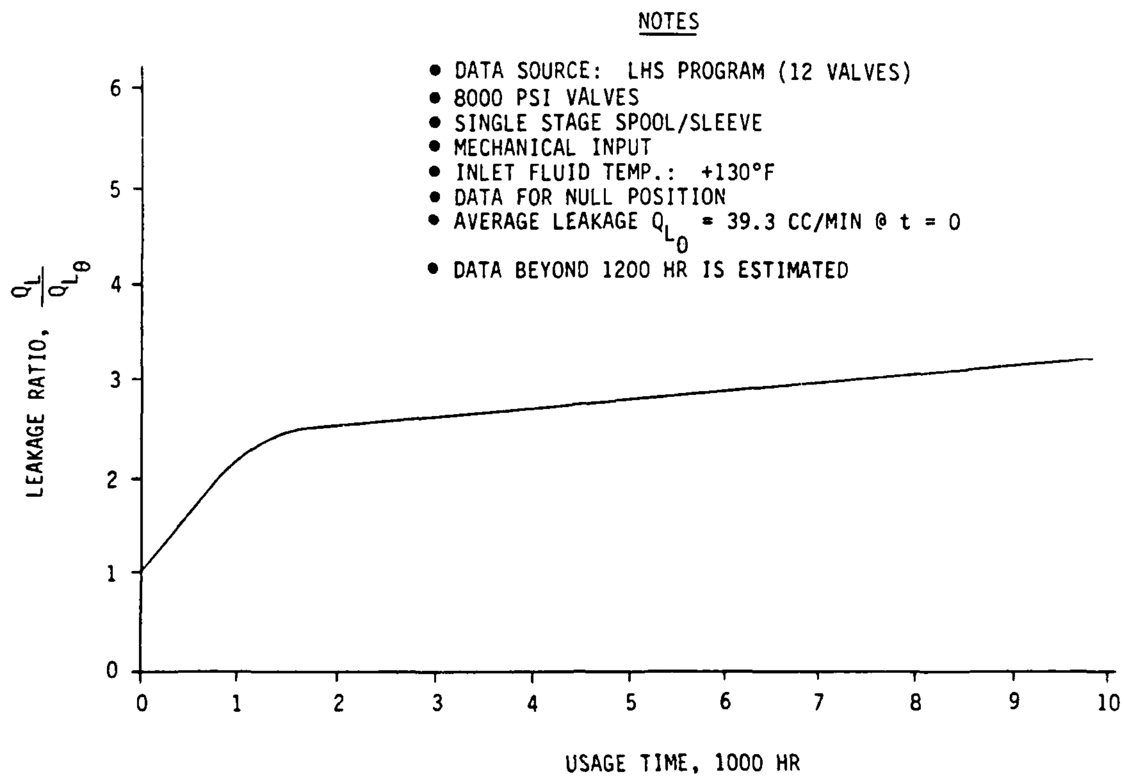


Figure B-21. Valve leakage vs. age

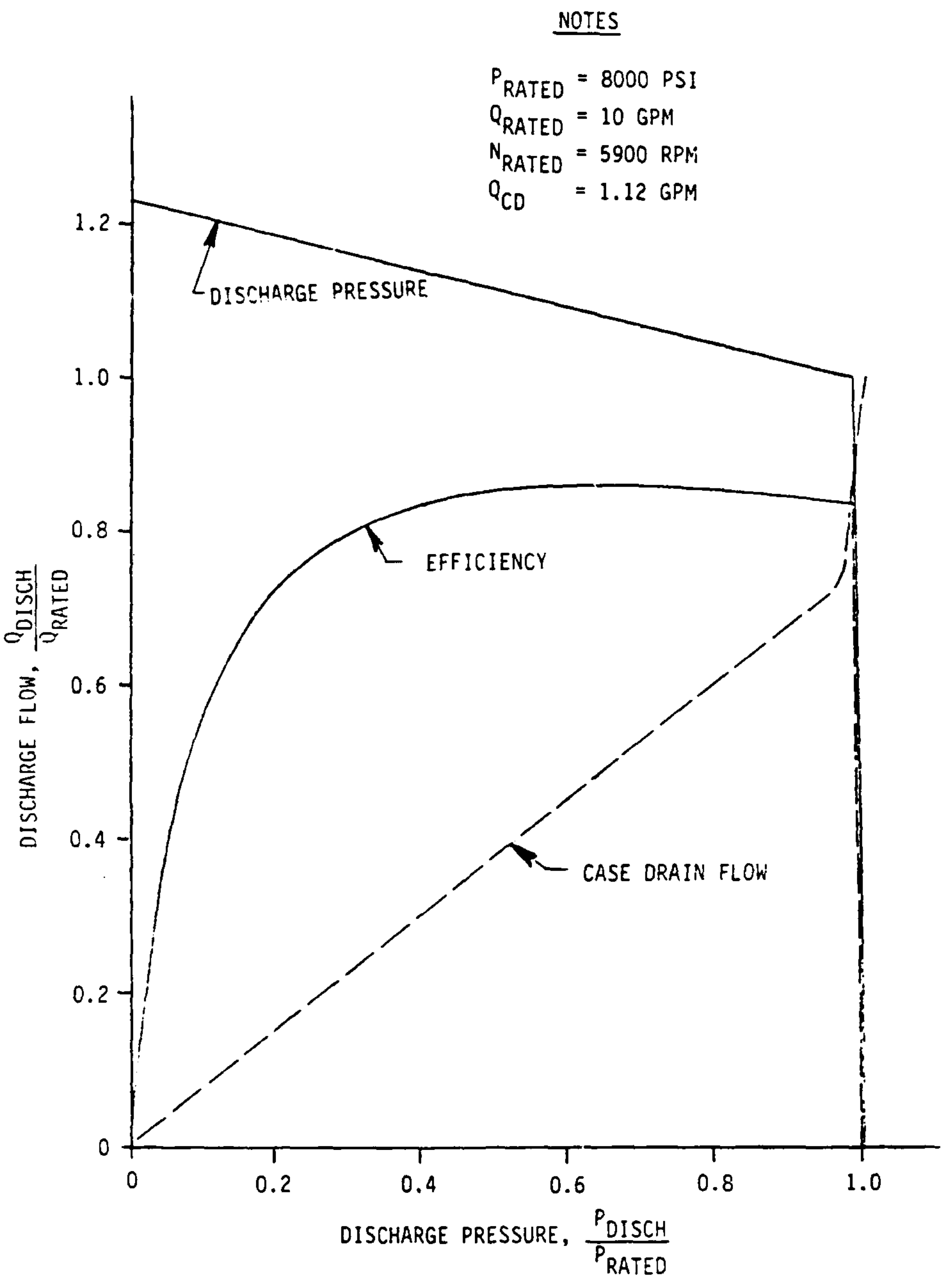


Figure B-22. Baseline pump performance characteristics

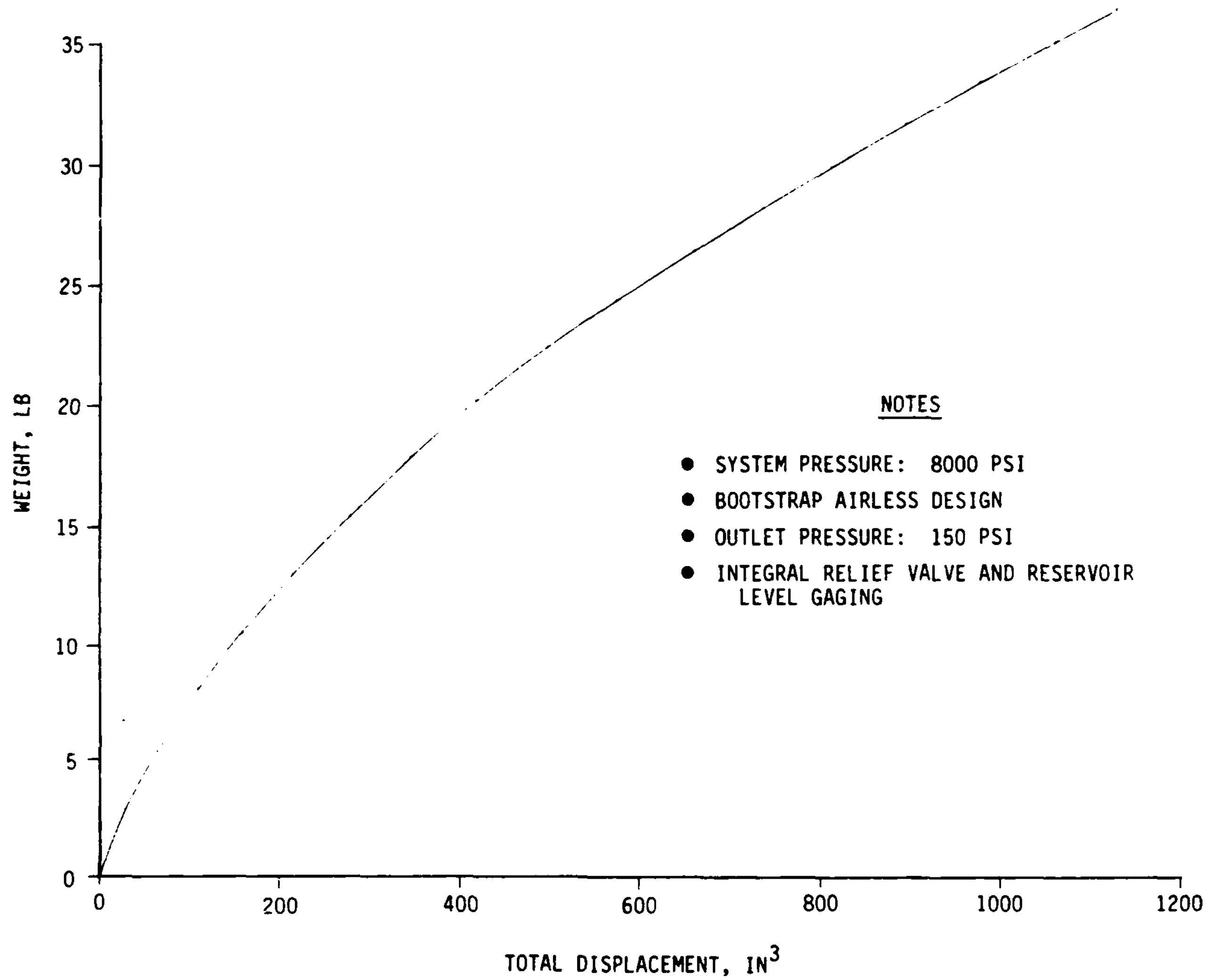


Figure B-23. Reservoir weight

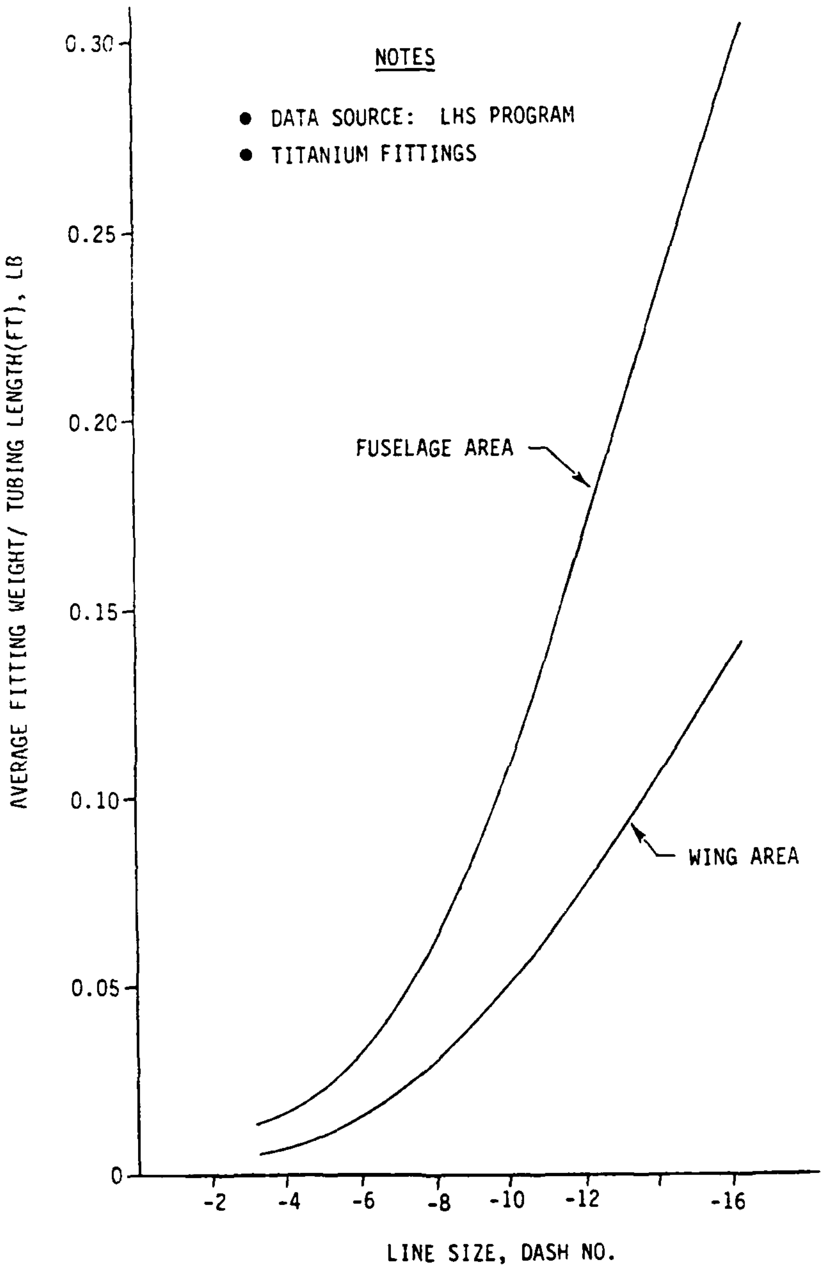


Figure B-24. Average fitting weight per foot

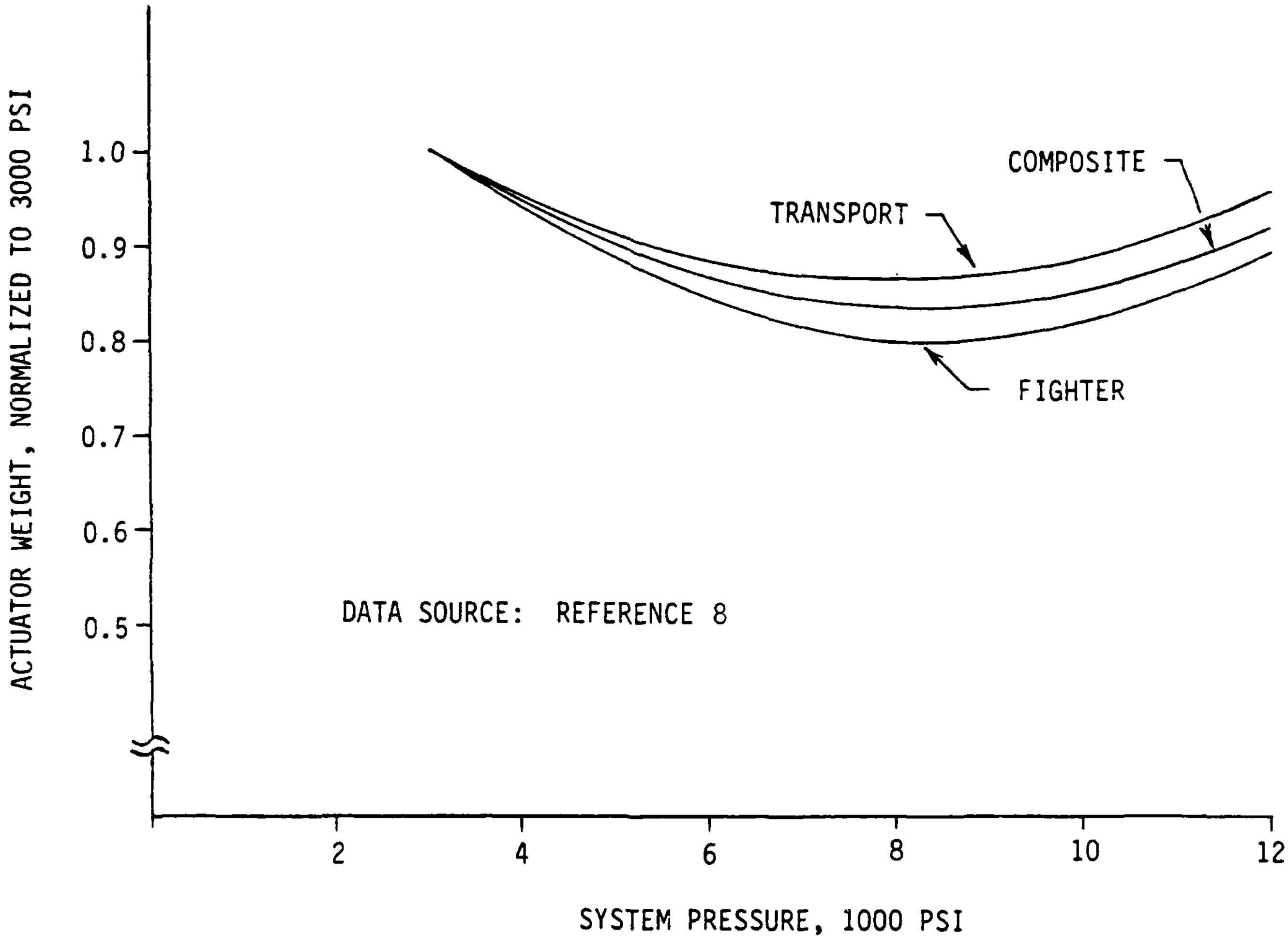


Figure B-25. Actuator weight vs. system pressure

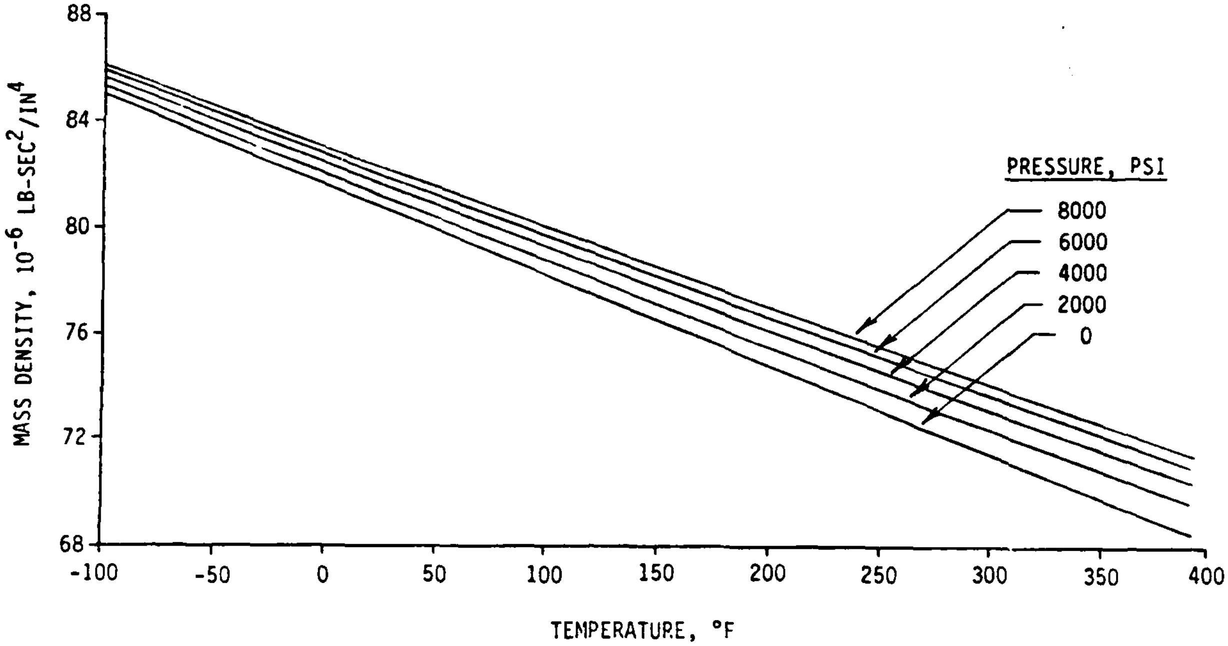


Figure B-26. Density, MIL-H-83282 hydraulic fluid

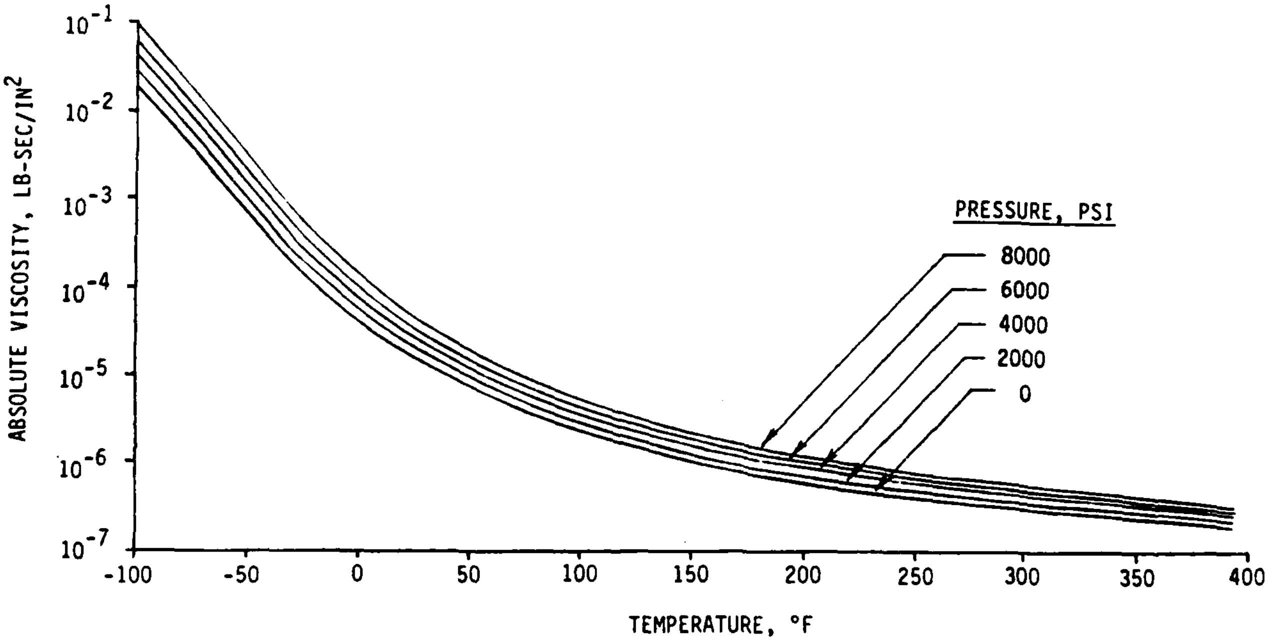


Figure B-27. Viscosity, MIL-H-83282 hydraulic fluid

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